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This Month's Cover: Gisholt hydraulic automatic lathe. All control valves are actuated by pressure variation within the circuit. Front carriage has standard tool blocks and the rear slide has a universal tool block to allow fast and simple tooling change-overs.

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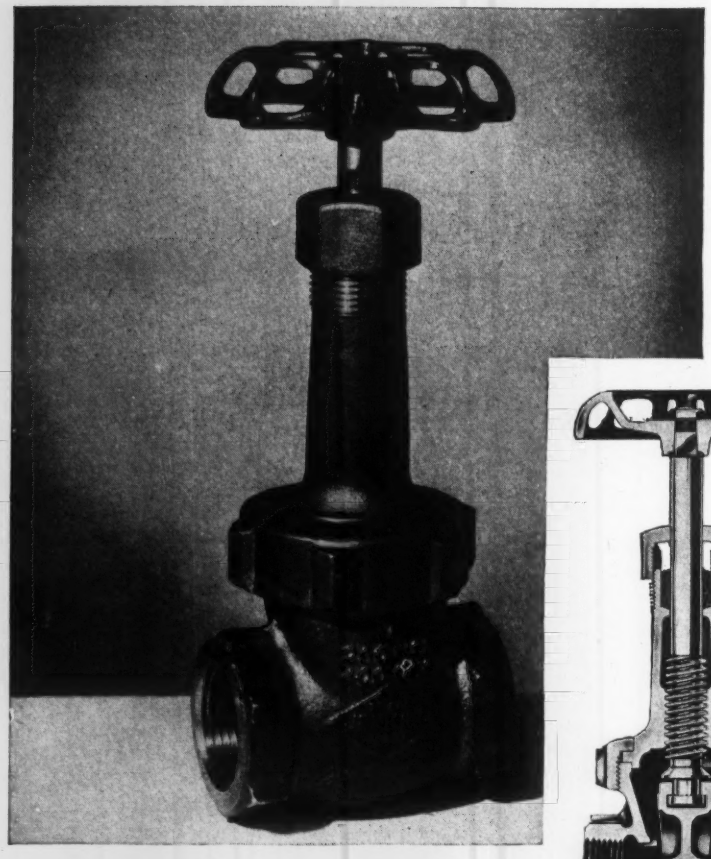

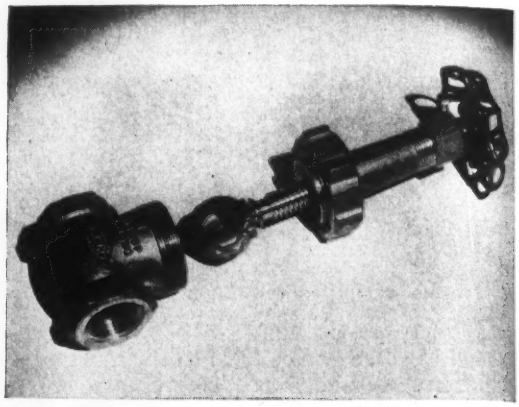

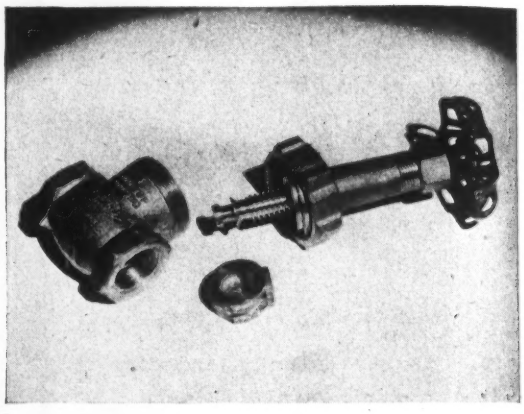

Above views show how easy it is to replace the bronze wedge. New wedges are inexpensive and readily available. The valves, made by Jenkins Bros., 80 White Street, New York 13, are now carried in stock by all regular suppliers of Jenkins valves.

Cut-away view of the new Jenkins Fig. 270-U bronze gate valve. It is the first time the combination of a bronze wedge and Monel seat rings has been available in a small, moderately-priced valve. The seat rings are worked to a hardness of 190 Brinell—about 2½ times as hard as valve bronze.

NEW VALVE

solves old maintenance problem

The wedge of a gate valve is much easier to replace than the seats. When the valve starts to leak, all a workman has to do is shut off the pressure, unscrew the bonnet and replace the wedge.

Unfortunately, the seats usually take more of a beating than the wedge. When the valve is open, the wedge is up out of the stream flow. The seats, on the other hand, are exposed to the erosive and corrosive action of whatever fluid the pipe line is carrying.

This gave designers at Jenkins Bros. an idea. Why not develop a small, moderately-priced gate valve in which the seats could be counted on to outlast the wedge? It sounded easy... just make the seats out of a more durable material than the wedge. If they could find a hard, corrosion-resistant material that was also machinable, they could substitute wrought seat rings for the cast ones found in more costly valves. Monel furnished them with the answer.

The new valve is the Jenkins Fig. 270-U bronze gate valve, available in sizes from ¼ inch to 2 inches. It will handle 200 lbs. of 500°F. steam or 400 lbs. of oil, water or gas.

The wrought Monel seat rings are expanded into the valve body. In tests, the high-quality bronze wedge seated tightly against them through several thousand openings and closures carrying steam at normal working pressures. When it finally wore, it was replaced, and the tight seal was restored. A new wedge for a 1-inch valve costs about $1.00. Thus, at a maximum cost of a few dollars, the service life of the new valve can be indefinitely prolonged—larger Monel-trimmed gate valves have lasted 25 years or more.

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Topics

MELTING NOSES are accompanied by freezing tails in missiles flown at thousands of miles an hour at great altitudes. Overheating of the nose of aircraft or missiles is one of the many problems of designers in this supersonic era. The fact that the tails become cooler as noses become hotter suggests the possibility of an internal cooling system.

PRECISION CASTING method for various metals, developed by the Croning company, Hamburg, Germany, uses a molding material consisting of a dry mixture of high-grade silica sand and a plastic binder in powder form. Castings of iron, steel and aluminum have unusually smooth and clean surfaces in the as-cast condition.

PHOTOELASTIC RESINS in the Fosterite family can now be cast in large sections for three-dimensional studies of "frozen stresses". Developed experimentally at Westinghouse Research Laboratories, the resin has been cast in a six-inch diameter and three-foot long piece. In the past models have been limited to 1⅛ inches in thickness.

EXPERIMENTAL MOTOR, running at a speed of 204,000 rpm, has been built by the Bryant Chucking Grinder Co. The unit is a 3-phase induction type motor operating on 3,400-cycle current.

GASOLINE ADDITIVES, no matter what the substance, when added to gasoline in the amounts of one per cent or less will not increase measurably gasoline mileage, the ease of starting and warm-up, or the power if knock is not present in an automotive engine.

COLOR MICROSCOPY process developed at Bausch & Lomb Optical Co. provides a speedy, accurate identification of a wide range of colorless transparent substances, including minerals and ores. Light from the microscope lamp is passed through a dark-field substage lens to strike the sample at a high angle. A high dispersion liquid covers the sample and matches the light-bending ability of the materials in the sample at different portions of the color spectrum, producing highly colored material identification.

STANDARD OF LENGTH, superior to the meter bar of platinum-iridium, exists in the wavelength of green radiation of mercury 198. Preliminary measurements at the National Bureau of Standards have shown an accuracy of one part in one hundred million. One part in a billion is theoretically possible.

INSULATING MATERIAL, composed of purified asbestos fibers mixed with a mineral binder and filler, is produced in paper-like sheets of one mil thickness. Known as Terratex, the sheet has a tensile strength of 4000 pounds per inch when treated with varnish. Its unusual dielectric strength, thermal life and incombustibility fill a gap for designers in the field of high-temperature electrical insulations.

BREAK-STEP OPERATION of looms in textile mills has been found desirable to reduce the vibration and stress that would otherwise be set up in the machine members. In a Westinghouse control for out-of-step operation, each loom in a pair has two limit switches and two magnetic relays mounted in a common control box, plus a mechanical cam arrangement driven by the crankshaft to trip the switches.

RESEARCH is gaining the attention it deserves. According to a recent survey 87 per cent of American industry is spending more money for research than it did before the war and 72 per cent anticipate an increase in research activities. Of these activities 15 per cent in point of time is being devoted to new products and processes in fields that are foreign to the organization's main efforts.

Which Wrought Stainless Steel?

By Stanley P. Watkins
Manager

and

Roland J. Berkol
Engineer

Development Engineering Dept.
Rustless Division
The American Rolling Mill Co.
Baltimore, Md.

Part I–Properties and Processability

STAINLESS steel, bright and rustless, is today a "proved" metal. More than twenty years have passed since the days when, as a "new" metal, it was struggling to justify its higher initial cost. Strangely enough, stainless today is one of the few stabilizing influences in our inflation-bound economy, its price having increased only 10 per cent since 1940, compared to 54-to-58 per cent for less noble competitive metals.

Fig. 1—Top—Some thirty standard types of stainless offer properties to suit the requirements of an almost numberless variety of machine parts

The strength and ductility of stainless sheets and strip give designers an opportunity to create new and more efficient structures with higher unit stresses and lighter sections. Heat resistance of certain stainless grades overcomes high-temperature problems. Bars and wire are available in a wide variety of shapes, mechanical conditions and finishes. Besides the standard grades, special compositions are produced to meet unusual requirements of fabrication and use. These include the free-machining grades. Free-cutting elements and special mill processing make Types 416 and 430-F machinable at speeds approaching 85 per cent of those used on Bessemer screw stock. Many grades are hardenable by heat

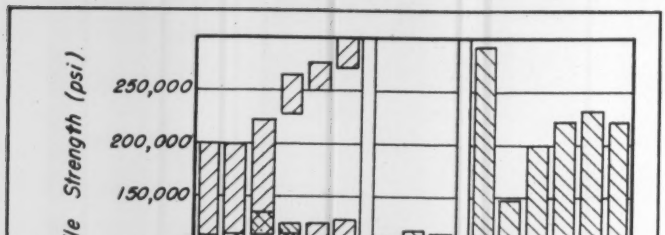
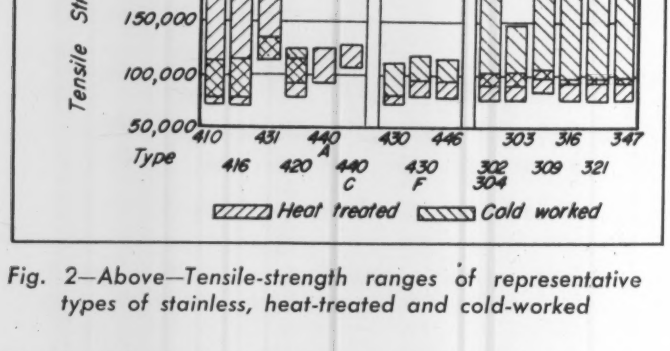


Fig. 2—Above—Tensile-strength ranges of representative types of stainless, heat-treated and cold-worked

Fig. 3—Below—Yield-strength ranges of representative types of stainless in heat-treated and cold-worked conditions



treatment and offer excellent mechanical properties.

There is no rule-of-thumb method for selecting the right grades of stainless steel for various uses and products, such as are shown in *Fig.* 1. Characteristics of each grade should be considered after determining how the steel will be (1) finally used, and (2) fabricated. Designers are urged to consult with stainless mill engineers and metallurgists when there is any doubt about the proper grade to be used. The tables and charts presented in this article give a general idea of the variety of properties available.

In order to simplify grade selection and standardize the various accepted compositions, the American Iron and Steel Institute has assigned type numbers. From TABLES I, II and III, it will be seen there are about 30 standard types. With the exception of the free-machining (Types 303, 416, 420F, 430F and 440F) and high-carbon (Types 420, 440A, 440B and 440C) grades, most are available in sheets, strip, plates, bars and wire. In the excepted categories flat-rolled forms generally are not produced. Still other grades are regularly made by some stainless producers, and many more may be found in various stages of research and development to meet the special requirements of new future applications.

PROPERTIES: Wrought stainless steels offer: (1) Remarkable resistance to a wide variety of corrosion conditions, no plating to peel or wear off, no paint to chip, great dimensional stability and resistance to abrasion and corrosion; (2) twice the stiffness of copper-base alloys, three times that of aluminum, five times that of magnesium, and many times that of plastics; (3) toughness that will absorb great shock loads even at extremely low temperatures and exceptional endurance strength; and (4) good overall economy in spite of somewhat higher initial cost compared with carbon steels because stainless steels insure long life with low maintenance. The design engineer's freedom of choice is indicated by the charts, *Figs.* 2 to 8.

Tensile Strength and Yield Strength: Even in the annealed condition stainless steels are decidedly strong. The tensile and yield strengths of the chromium-nickel types cannot be raised by heat treatment, but can be increased to high values through cold-working. Similarly, the ferritic chromium grades can be increased in strength only by cold-working, but to a much lesser extent. On the other hand, the martensitic chromium types may readily be given high-strength values through heat-treatment. *Figs.* 2 and 3 give in bar-chart form the tensile and yield strength ranges. In the cases of 420, 440A and 440C, the top ranges apply to the alloys as hardened, while the lower ranges apply to the annealed or tempered conditions.

Elongation and Reduction of Area: The amount of stretching to which a stainless steel may be subjected before fracture is a measure of its ductility and is of particular interest in the selection of grades for deep drawing. Reduction of area, another indication of ductility, is used only on bars and large wire. It is a measure of the degree to which the test sample necks-down at the point of fracture, being the per cent decrease in cross-sectional area from the diameter of the specimen to the diameter of the fractured sample. Typical elongation and reduction of area (R. A.) values are shown in TABLE IV.

Izod Impact Strength: This is at best a crude measure of the likely toughness of a sharply notched section. For comparative purposes it may be useful, however, and is a measure of the amount of energy (in foot pounds) absorbed in fracturing a notched specimen. Results are affected by specimen size,

TABLE I

Chromium-Nickel Types of Stainless

(Austenitic; Non-hardenable; Non-magnetic)

A.I.S.I. Type No.	Carbon	Chromium	Nickel	Other Elements
301	0.08-0.20	16.00-18.00	6.00- 8.00	...
302	0.08-0.20	17.00-19.00	8.00-10.00	...
302B	0.08-0.20	17.00-19.00	9.00-10.00	Si 2.00-3.00
303	0.15 max.	17.00-19.00	8.00-10.00	P or S or Se 0.07 min, Zr or Mo 0.60 max.
304	0.08 max.	18.00-20.00	8.00-11.00	...
305	0.12 max.	17.00-19.00	10.00-13.00	...
308	0.08 max.	19.00-21.00	10.00-12.00	...
309	0.20 max.	22.00-24.00	12.00-15.00	...
310	0.25 max.	24.00-26.00	19.00-22.00	Si 1.50 max.
314	0.25 max.	23.00-26.00	19.00-22.00	Si 1.50-3.00
316	0.10 max.	16.00-18.00	10.00-14.00	Mo 2.00-3.00
317	0.10 max.	18.00-20.00	11.00-14.00	Mo 3.00-4.00
321	0.08 max.	17.00-19.00	8.00-11.00	Ti 5 x C min.
347	0.08 max.	17.00-19.00	9.00-12.00	Cb 10 x C min.

All types contain 2.00 per cent maximum Mn and, unless otherwise shown, 1.00 per cent maximum Si, 0.04 per cent maximum P, and 0.03 per cent maximum S.

depth and sharpness of notch. *Fig.* 4 gives relative values obtainable in the tempered or annealed vs. hardened and stress-relieved conditions.

Creep Strength: In many high-temperature applications the designer must know the maximum stress which, when applied at a certain temperature, will then reuslt in a known rate of elongation in a certain period of time. This stress, known as "creep strength", usually is expressed as the stress in pounds per square inch of cross section that will cause an elongation of 1 per cent of original length in 10,000 hours. *Fig.* 5 shows the comparative creep strengths of various types of stainless steel at 1000 and 1200 F.

Thermal Expansion and Thermal Conductivity: These characteristics, familiar to designers, are covered in bar-chart form by *Figs.* 6 and 7, which are self-explanatory.

Scaling Resistance: Stainless steels are widely used in the "hot" jobs because of their superior strength and resistance to destructive scaling at elevated temperatures. They owe their scaling resistance to the protection of the highly impervious and adherent chromium-oxide film. This oxide layer is formed on exposure to high temperatures and within its effective range, protects the metal from rapid deterioration.

The long-time protective ability of this oxide layer depends directly upon the amount of chromium in the steel; that is, the higher the percentage of chromium, the greater the resistance to scaling. Since stainless steels have different specified chromium contents, they also have different temperature levels below which they are safe from destructive scaling. Scaling is accelerated in applications where the protective oxide layer is constantly being broken. Repeated heating and cooling, with alternate expansion and contraction of the metal, will crack the protective oxide layer and cause it to flake off. Under such conditions the chromium stainless steels usually tend to scale less than the chromium-nickel grades because of their lower thermal expansion.

In high-temperature applications scaling generally is caused by oxidation and accelerated through chemical attack by the various products of combustion. *Fig.* 8 shows the destructive scaling or safe maximum working temperatures of representative types of stainless steels when employed under average conditions.

TABLE II

Hardenable Chromium Types of Stainless

(*Martensitic; Magnetic*)

A.I.S.I. Type No.	Carbon	Chromium	Nickel	Other Elements
403	0.15 max.	11.50-13.00	...	Si 0.50 max.
410	0.15 max.	11.50-13.50	...	...
414	0.15 max.	11.50-13.50	1.25-2.50	...
416	0.15 max.	12.00-14.00	...	Mn 1.25 max., P or S or Se 0.07 min., Zr or Mo 0.60 max.
420	Over 0.15	12.00-14.00	...	...
420F	Over 0.15	12.00-14.00	...	P or S or Se 0.07 min., Zr or Mo 0.60 max.
431	0.20 max.	15.00-17.00	1.25-2.50	...
440A	0.60-0.75	16.00-18.00	...	Mo 0.75 max.
440B	0.75-0.95	16.00-18.00	...	Mo 0.75 max.
440C	0.95-1.20	16.00-18.00	...	Mo 0.75 max.
440F	0.95-1.20	16.00-18.00	...	P or S or Se 0.07 min., Zr or Mo 0.60 max.

Unless otherwise shown, all types contain 1.00 per cent maximum Mn, 1.00 per cent maximum Si, 0.04 per cent maximum P, and 0.03 per cent maximum S.

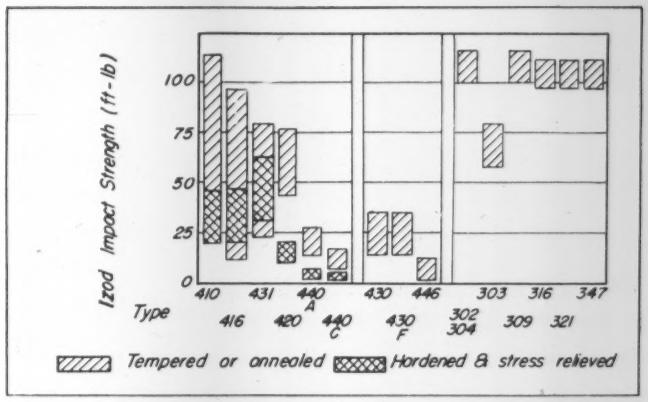
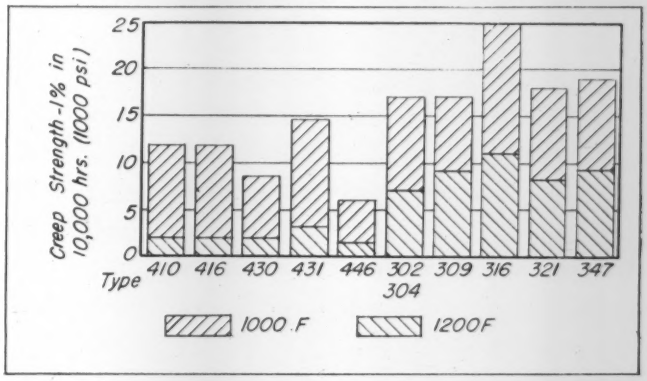


Fig. 4—Above—Impact-strength ranges of representative types of stainless, annealed and hardened by heat treat

Fig. 5—Below—Creep strengths of representative stainless types at temperatures of 1000 F and 1200 F



Electrical Resistivity: Specific electrical resistivity of the several grades is shown in TABLE V.

Tensile Strength at Elevated Temperatures: Tensile properties of carbon steel generally are unsatisfactory at temperatures exceeding 900 to 950 F. Yet certain types of stainless steel, such as 309 and 310, are used at temperatures up to 2000 F, not only because of less scaling but because of their relatively high tensile strength at elevated temperatures. *Fig.* 9 shows the short-time, high-temperature tensile properties of various types of stainless steel.

Subzero Service: One of the many unusual properties of the chromium-nickel stainless steels is illustrated in *Fig.* 10. From the equivalent of desert to substratosphere temperatures, these grades actually exhibit slightly increased impact strength and retain much of their excellent ductility. Cold-drawn material of 250,000-psi tensile strength at room temperature has 40 ft-lbs Izod impact strength at −80 F. All the way down to −400 F the chromium-nickel stainless steels out-perform many other materials.

Corrosion Resistance: Each type of stainless steel reacts somewhat differently to corrosive media. While in some cases the success of a stainless application may be confidently predicted, under actual service conditions impurities in chemicals and chemical compounds frequently alter results. Some impurities in-

TABLE III

Nonhardenable Chromium Types of Stainless

(*Ferritic; Magnetic*)

A.I.S.I. Type No.	Carbon,	Chromium	Nickel	Other Elements
405	0.08 max.	11.50-13.50	...	Al 0.10-0.30
406	0.15 max.	12.00-14.00	..	Al 3.50-4.50
430	0.12 max.	14.00-18.00	...	...
430F	0.12 max.	14.00-18.00	...	Mn 1.25 max.; P or S or Se 0.07 min.; Zr or Mo 0.60 max.
442	0.25 max.	18.00-23.00	...	...
446	0.35 max.	23.00-27.00	0.25 max.	Mn 1.50 max.,

Unless otherwise shown, all types contain 1.00 per cent maximum Mn, 1.00 per cent maximum Si, 0.04 per cent maximum P, and 0.03 per cent maximum S.

hibit corrosion while others accelerate it.

Tables indicating the relative resistance of representative grades to various corrosive media are available from reliable stainless steel producers; space limitations do not permit their reproduction here. Such information should not be used as a positive guide for tendency toward pitting or localized attack. Practically all the halogen (chlorine, bromine, fluorine, and iodine) salts tend to pit stainless. The degree of pitting depends upon actual service conditions such as concentration, temperature, degree of aeration and duration of contact. For example, Type 302 is used regularly and satisfactorily for handling bleaching solutions such as sodium and calcium hypochlorite, even though under certain conditions these solutions would severly pit Type 316, which in general is the most resistant to pitting of all the stainless grades.

In general, three types of stainless steel, namely 302, 316 and 430, will indicate the average stainless qualities of other related grades. A grouping under the three types may be made:

Type	Related Grades
302	301, 303, 304, 308, 321 and 347
316	317
430	430F, 440A, 440B and 440C

As a general rule the higher the content of chromium and nickel, the greater the corrosion resistance. Thus greater corrosion resistance may be expected from Type 309 than from Type 302. However, for corrosion resistance comparable with Types 410 and 430, the high-carbon chromium grades must be in the fully hardened condition.

FABRICATION CHARACTERISTICS: Stainless-steel bar wire is more resistant to cold deformation than carbon steels, and more energy is required to shape it. However, it is readily cold-drawn, cold-headed and swaged when proper practices are used. The chromium grades do not harden appreciably when cold-worked, but the chromium-nickel grades work-harden rapidly.

Drawing: Chromium-nickel types of stainless steel are the popular drawing grades. They have ductility even greater than that of deep-drawing carbon steel. It should be remembered, however, that the chromium-nickel types all work-harden rapidly. Slower press speeds are necessary for maximum depth draws to permit more uniform work-hardening and elongation. Also, these grades are exceptionally strong, so greater power is required to complete a draw.

Chromium stainless steels are much less ductile than the chromium-nickel grades. In fact, they have only about three-fourths the ductility of deep-drawing carbon steel and about the same work-hardening chracteristics. Neverthless, they can be satisfactorily deep drawn although they may require more intermediate annealing. Ductility of the chromium grades may be improved by heating blanks in boiling water prior to drawing.

Forming: Stainless-steel sheets, strip and plates are brake and roll-formed with the same methods and procedures used for mild steel. However, considerably more power is required and greater allowance must be made for springback.

Machining: Owing to their great strength and toughness, even in the fully annealed condition, ordinary grades of stainless steel are limited to about one-half the machining speeds of Bessemer screw stock. To raise machinability to approximately 85 per cent that of Bessemer screw stock, the free machining grades Types 416 and 430F are available. Type 303 is an improvement over the regular "18-8" types; types 420F and 440F over their companion high-carbon chromium grades. Machining practices for the several compositions are slightly different.

Fig. 6—Thermal expansion coefficients of representative types of stainless within range of 32 to 1000 F

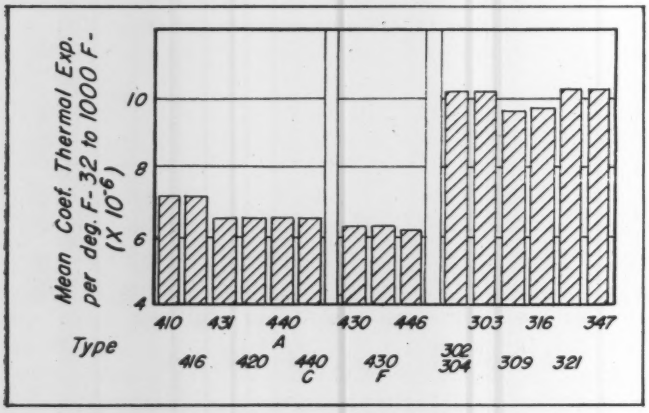


Fig. 7—Wide range of thermal conductivities is available from the many types of stainless

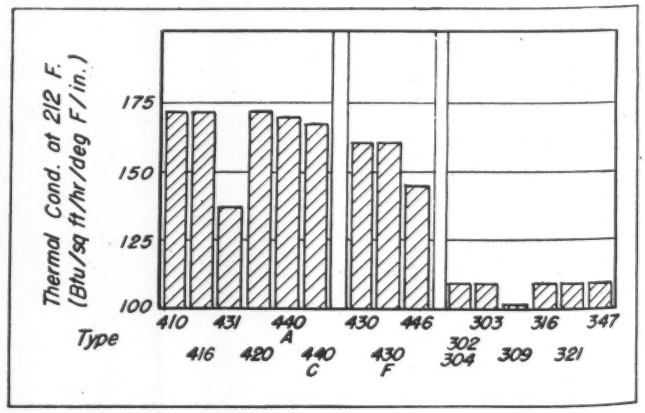


Fig. 11 shows the comparative recommended cutting speeds for various types of stainless steel.

Spinning: Greater power is required to spin stainless steel because of its higher strength; speed should be reduced to one-half or one-third of that used for softer metals and more frequent process annealing may be necessary than with ordinary spinning steel.

Torch-Cutting: Economical cutting of stainless-steel plates and heavy sections was made possible by a recently perfected flux-injection oxyacetylene method. Standard cutting equipment is provided with a modified cutting torch that delivers a flux to the torch tip. This flux combines chemically with the protective stainless oxides, which otherwise would act as insulation, to produce a low melting-point, free-flowing slag. By this method stainless steel is torch-cut, after a normal preheat, at about the same speed as carbon steel.

Hot Forging: Forging characteristics of a stainless steel depend upon its analysis. Selection of a suitable grade and proper control of forging temperatures insure success. Wrought stainless steels are readily forged into complicated shapes.

Welding: Practically all wrought stainless steels can be welded readily by the electric arc, oxyacetylene, atomic hydrogen, helium arc, and resistance methods. Only the high-carbon and free-machining types require special procedures. The chromium-nickel grades weld with particular smoothness and have excellent ductility in the welded condition. Heavy sections that cannot be annealed after welding, or welded parts subject to corrosive conditions or high-temperature gases, generally are made from Type 321 or Type 347.

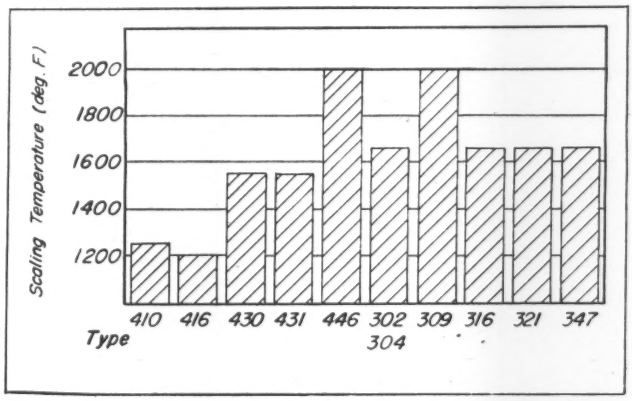


Fig. 8—Above—Maximum temperatures at which various types of stainless can be used without danger of scaling

Fig. 9—Below—Some types of stainless offer relatively high tensile strengths at elevated temperatures

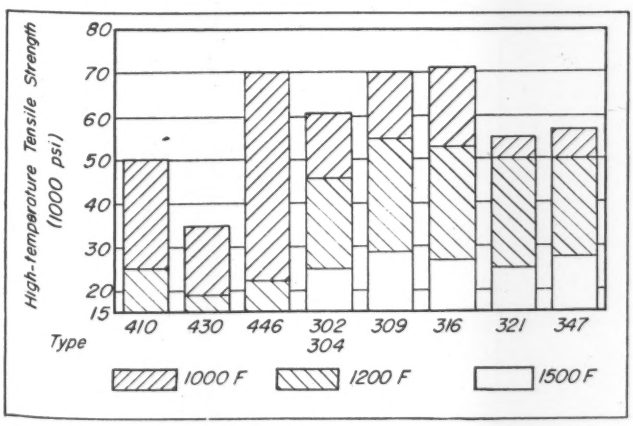


TABLE IV

Typical Elongation and Reduction-of-Area Values Of Stainless Steels

Condition	Wire Elong. (%)	Wire R.A. (%)	Bar Elong. (%)	Bar R.A. (%)
CHROMIUM-NICKEL TYPES				
Annealed	20-60	60-70	45-60	55-70
Cold-drawn	15-55	45-55	25-45	50-65
FERRITIC-CHROMIUM TYPES				
Annealed	10-35	50-70	25-30	55-65
Cold-drawn	....	....	15-20	55-65
MARTENSITIC-CHROMIUM TYPES				
Annealed	5-30	15-70	12-35	20-70
Cold-drawn	10-15	50-60	5-17	15-60
Heat-treated	5-10	40-50	14-17	40-50

Soldering: Stainless steels are soldered by ordinary methods. Even so, fluxes must be stronger than those used for mild steel to cut through the resistive chromium-oxide film. Longer application of heat is necessary since stainless conducts heat more slowly than carbon or alloy steels.

Riveting: Stainless-steel rivets may be cold or hot-driven with conventional equipment, but work hardening, air hardening and grain growth must be given due consideration. Chromium-nickel grades of rivets are preferable for all-riveted assemblies.

HEAT TREATING AND PICKLING: In the fabrication of stainless steels, heat-treatment often is necessary to anneal, harden, temper or stress-relieve. When heating, impinging of flames on the work should be avoided. Muffle, controlled-atmosphere, or salt-bath type furnaces are recommended for this reason.

To make the most of the excellent corrosion resistance and mechanical properties of stainless steels, recommended heat-treating procedures should be followed. These are known for every stainless type and the information can be readily obtained. Where service demands a hard and wear-resistant surface, stainless steels can be surface-hardened by special nitriding methods commercially available.

Scale formed on stainless steels by annealing and

Fig. 10—The chromium-nickel types of stainless exhibit excellent ductility at temperatures far below zero

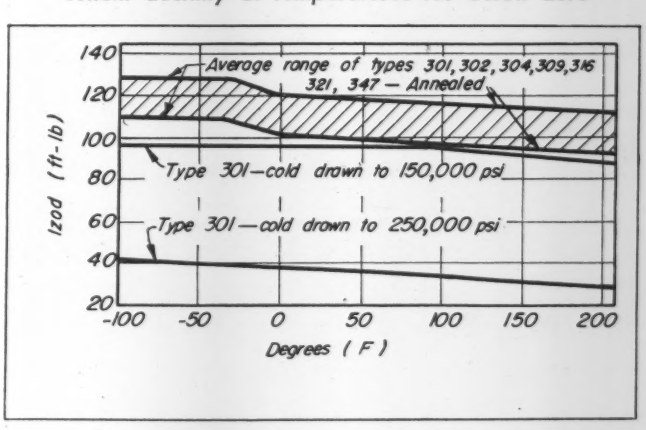


TABLE V

Electrical Resistivity of Stainless Steels

(*microhms/sq cm/cm*)

Stainless Types	Spec. Elec. Resis. at 70 F
420	55
403, 410 and 416	57
405, 430, 430F, 440A, 440B and 440C	60
442	64
446	67
414	70
301, 302, 302B, 303, 304, 308, 321 and 431	72
347	73
316	74
309 and 310	78

welding processes is sometimes tenacious and difficult to remove by conventional steel pickling procedure. There are several effective processes for the softening of the scale before removal. Final scale removal and whitening is accomplished in a solution containing 6 to 15 per cent nitric acid and ½ to 1½ per cent hydrofluoric acid.

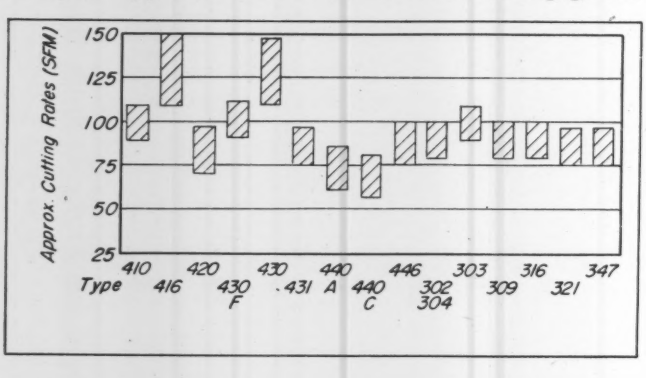
PASSIVATION: Passivation should be the final operation on a stainless-steel part to provide a chemically clean surface free of foreign substances. This may be accomplished by submerging the stainless steel part in a 20 to 40 per cent nitric acid solution at 120 to 140 F for a period of 20 to 30 minutes and then rinsing in clean running water and drying. For polished or hardened chromium parts a solution of 18-22 per cent nitric acid plus 1½ to 2½ per cent sodium dichromate at 110 to 120 F is recommended.

SURFACE FINISHING: Various methods of polishing and buffing are used to bring out the beauty and lustrous appeal of stainless steel. In standard practice a rough grind comes first with No. 60 or No. 80 grit on glue-headed solid cloth wheels or abrasive belts. Grinding proceeds with progressively finer grits through No. 120.

Polishing requires loose, resilient, soft fabric wheels to which aluminum oxide abrasive and lubricant are applied by holding a stick or cake of the combined abrasive and lubricant against the rotating wheel. Areas polished with progressively finer grits up to No. 320 present a surface suitable for buffing when this is desired. Buffing is done with soft muslin or canton flannel wheels and with special stainless buffing compounds which bring out the utmost luster, color and reflectivity.

There are some short-cuts to attractive finishes.

Fig. 11—Recommended cutting speeds for various types of stainless. Types 416 and 430F are free-machining grades



No. 2B finish on sheets and No. 2 finish on strip may be given a beautiful finish by Tampico brushing alone. The result is comparable with "Butler" finish on sterling or silverplate. Because of its dense, smooth surface, the 2B finish sheet may be polished and buffed without preliminary rough grinding. Intricate, hard-to-polish parts may be given a brilliant finish through economical electropolishing; while a special blackening process not only eliminates stainless brightness, when desired, but offers design possibilities as well.

Blackening: The blackening process adds a smooth, black oxide film, about one-hundred-thousandth of an inch thick to the surface of either chromium-nickel or chromium stainless steel. This blackened "skin" retains all the corrosion resistance of stainless steel; in fact laboratory tests indicate that the film actually improves the resistance of stainless steels to some weak acids. It is normally dull black in color but may be improved in luster, if desired, through the application of oils or waxes. The blackening shows no tendency to turn gray in service. Procedure involves the usual cleaning of the parts after which they are immersed in a molten bath of dichromates at a temperature of 730 to 750 F for from fifteen to thirty minutes. Following this, the parts are merely cooled and rinsed.

Blackened stainless steel parts are relatively new to the general consumer and many possibilities have not yet been developed. Among the many indicated uses are parts of cameras, motion picture projectors and sound equipment, as well as darkroom photograpic equipment.

Electropolishing: A brightening process, known as electropolishing, gives a high luster to stainless steel at costs much lower than mechanical polishing and buffing. It is the reverse of electroplating. In plating the work is the cathode and metal is added, while in electropolishing the work is the anode and metal is removed.

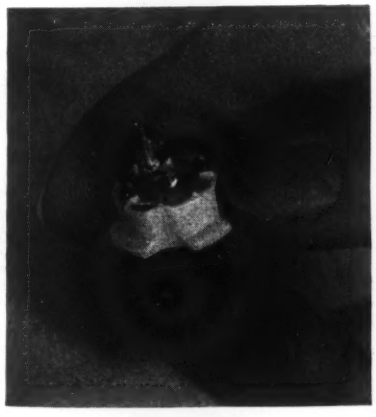
Commercial experience over a number of years has shown that many products, heretofore too costly or difficult to polish by buffing or tumbling, can now be economically and efficiently cleaned and brightened to an attractive finish. The electropolishing process may be applied to regular or irregular parts formed or drawn from sheets or strip, intricately formed wire products, resistance-welded wire products (also removes spot weld discoloration), parts too small to hold against wheels (also removes slight burrs), deeply recessed and complex parts, and forgings.

In comparison with mechanical polishing, metal loss is extremely low—from 0.0002 to 0.001 inch—depending upon the surface being electropolished. Since brightness and regularity of finish depend upon uniform metal and metal removal, smooth, dense surfaces free from scratches and tool marks are necessary. Whenever possible, cold-finished stainless steel should be used in making articles to be electropolished.

Part II of this series will deal with the applicability of various wrought stainless steels to different machine parts. Case histories will be presented, tying in design requirements with properties and characteristics of the types used.

Scanning THE FIELD for Ideas

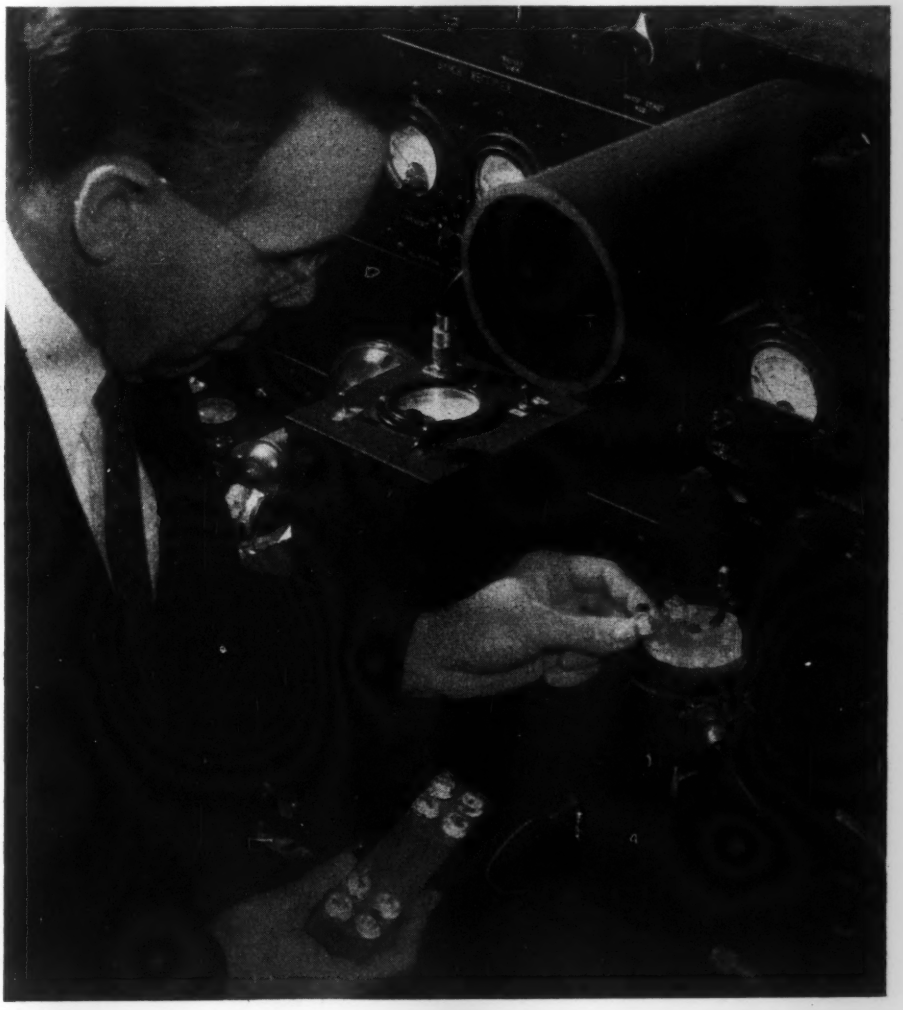
AMPLIFICATION of electric current by a radically new method discovered at Bell Telephone Laboratories may have far-reaching influence on the future of electronics. The method is based on the fact that when beams of electrons are shot at an insulator, in this case a diamond chip, electric currents are produced in the insulator which may be several hundred times as large as the current in the original electron beam. In some experiments electric currents shot at a diamond chip, such as shown below mounted in a ceramic tube



socket, have been amplified as much as 500 times, using this technique.

The diamond crystals used in the experiment, shown at right, are about 1/4-inch square and approximately 0.020-inch thick. Gold is evaporated onto the two flat surfaces of the chip in films less than 0.00001-inch to afford electrical connections. To prevent electrons from being trapped in the diamond by the tiny imperfections present in all crystals, a 120-cycle alternating-current is applied to the diamond. In this way the trapped charges are cancelled out and the induced current is allowed to flow freely. In bombarding the chips, successive pulses lasting only a millionth of a second and having energies of approximately 15,000 electron volts are used.

An important feature of the new technique is that

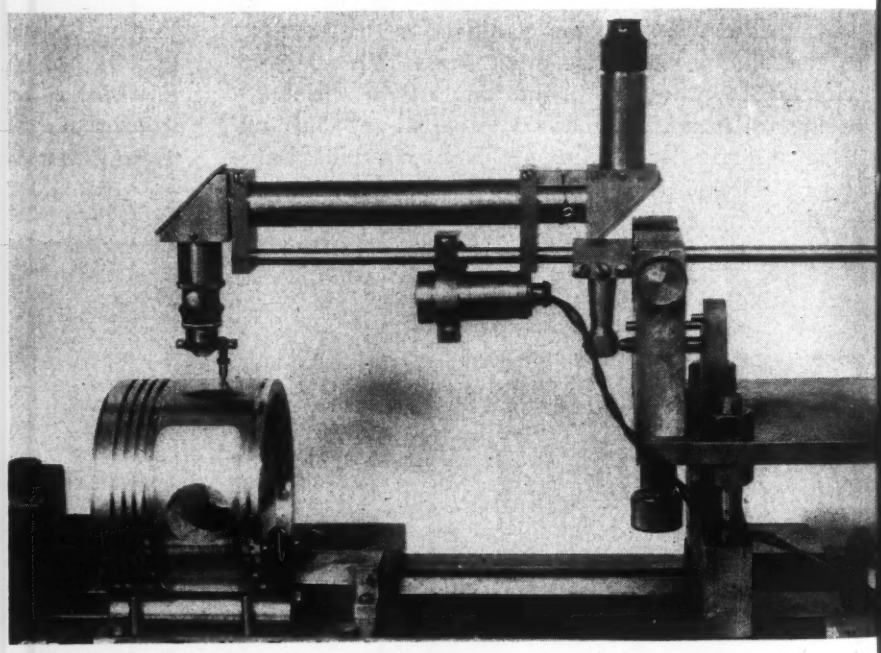


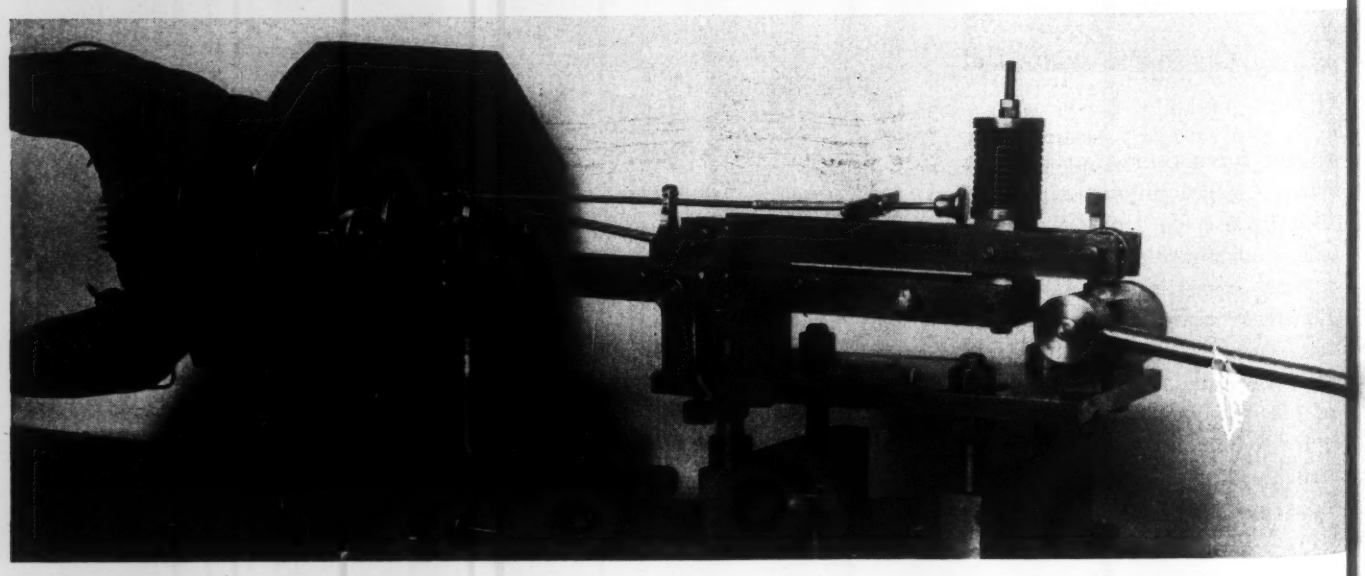
induced currents are produced within exceedingly short times. It is believed that the time is less than one ten-millionth second.

MEASUREMENT OF WEAR on machinery surfaces is facilitated by an improved indentation method developed at the National Bureau of Standards. By applying diamond-shaped markings to the working surfaces, wear as little as 0.00001-inch may be measured under favorable conditions. The wear is determined by carefully measuring the length of the indention before and after the test. Shown below is a photograph after test of a typical indentation mark on an engine cylinder. Originally the mark was about 0.9-millimeter long, the scale representing an actual length of 1 millimeter. For the diamond used, the ratio of depth of mark to length is about 1 to 35, thus one of the small scale divisions represents 0.0001-inch of wear.

Shown at bottom is an aircraft engine cylinder mounted in a fixture with the gage in position to make an indentation on the cylinder surface. To view and measure the diamond indentation before and after testing the apparatus illustrated below is utilized. It consists of a microscope and eyepiece scale, modified to be used as a periscope when viewing the inside of a cylinder. In this setup, however, it is being used to measure an indentation on a piston surface.

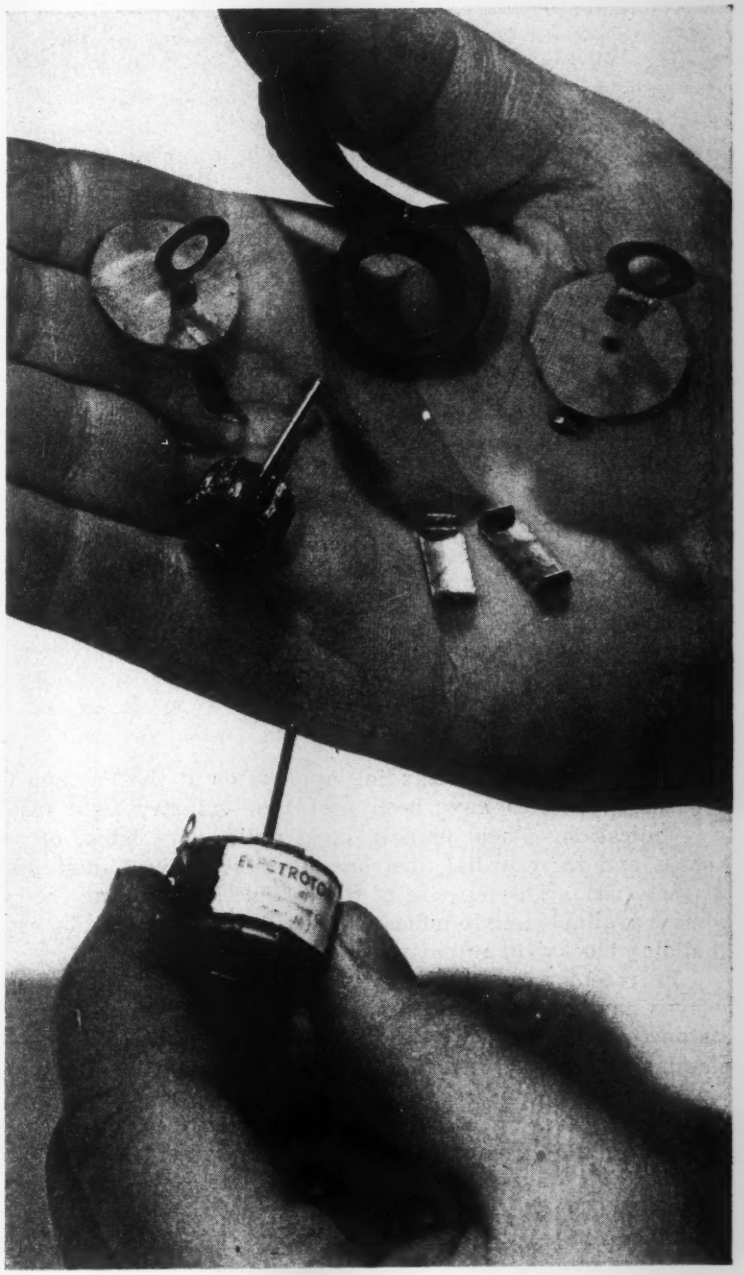
In using this method experience has indicated that results are not as satisfactory after the marks have been worn to about one-fourth of their original length or less. Greater accuracy is obtained by care-





fully regulating the size of the mark and adjusting the duration of the test, staying within the higher accuracy range of measurement.

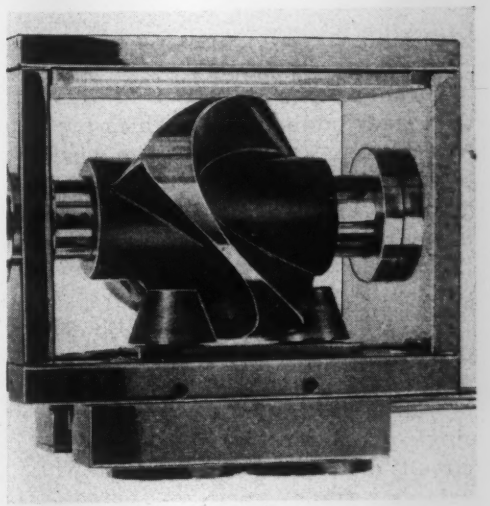
Indentations are made by impressing the apex of a four-sided diamond pyramid. From the standpoint of accuracy the chief limitation is that the sharp points of the marks are blunted somewhat when wear occurs. Possible use of cutting or grinding methods for producing marks without burrs and without sharp pointed ends presents a promising field for further investigation.



MIDGET MOTOR, right, utilizes a new type of rotor known as a "gap-ring armature." This unusual subminiature direct-current motor, weighing 3/4-ounce and developing 5000 rpm, dispenses with the usual separate commutator and segmentary windings. Motive power is derived from a permanent magnet and from an electromagnet operated from a dry-cell battery.

As shown in the disassembled view the gap-ring armature is mounted on a bobbin and spindle. Edges of the windings are bared so that they can complete an electrical circuit with contacts on the end plates. These windings have a core of insulated wires of high permeability upon which several layers of insulated copper wire are wound so that current flowing between the contacts energizes all the layers. The ring magnet serves as the frame and the motor is assembled with two clips which hold the laminated-plastic end plates.

The armature of this motor, designed by J. V. and J. E. G. Eurich of Lancashire, England, can be considered a horse-shoe electromagnet in which broken iron rings form the core and the outer winding the magnetic coil. The contact points on the armature, whose position relative to the armature gap in the core, determine the flux distribution and are, in effect, the brushes. This motor, to be manufactured by the Electrotor division of the United States Instrument Corp., has efficiencies comparable to other types of motors.



CAM DRIVE, right, for an internal-combustion engine is so designed that the pistons in the engine have true harmonic motion, while the drive shaft has a uniform rotary motion. In eliminating the conventional crankshaft the inventor, Vincent Palumbo, president of Pal-Vin Machine and Manufacturing Co., has designed an extremely compact engine. The number of pistons arranged around the cam are limited only by the size of the follower rollers on the cam.

Utilizing Selsyns in Machi

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SINCE their first outstanding application at the Panama Canal in 1914, selsyn devices have been useful in industry as a means of remote indication. Their uses in steel mills, wire mills, on machine tools, *Fig.* 1, and in recording instruments have utilized their self-synchronous characteristic. The purpose of this article is to outline the types of selsyn devices available and to indicate generally how they can be used as "building blocks" in automatic regulating systems.

In recent years, modified forms of conventional remote indicating selsyns have been used with amplidynes, electronic amplifiers, and in some instances with both. These systems are being used successfully on both metal and paper printing machines, paper machines, and wire stranding machines, as well as several forms of testing machines. Thus it has been possible to do away with large, troublesome mechanical linkages between various parts of a machine and at the same time to set up a self-regulating "closed cycle" system. Through combinations of the following types of selsyns it is possible to signal, control position, or regulate various functions of machines:

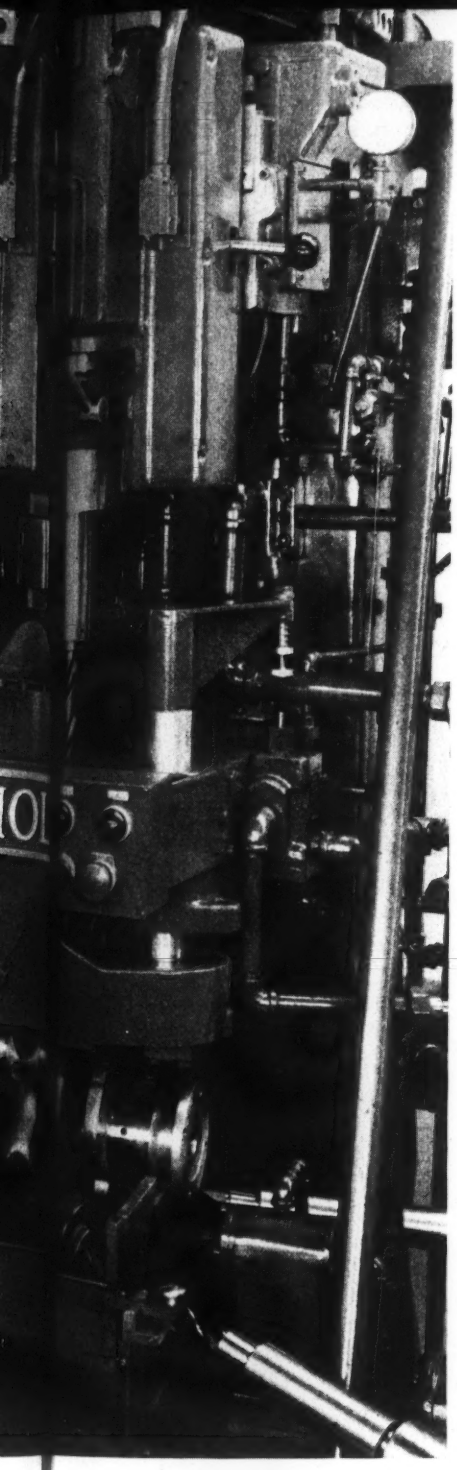
Transmitter (generator)
Receiver (motor)
Differential (generator or motor)
Voltage regulator
Control transformer

The selsyns discussed in this article should not be confused with the larger power selsyns which are similar in construction to wound-rotor induction motors, or with the small d-c instrument selsyns such as those used in airplanes. Extreme accuracy is not required in either of the foregoing kinds of selsyns. However, control and indicator selsyns, *Fig.* 2, have errors which may be as small as few tenths of a degree. Because of their high accuracy as control devices, they enjoy a much wider field of use than either of the other two types.

GENERATOR-MOTOR: In the simplest form of a selsyn system two units are used, one as a generator, shown in *Fig.* 3, and the other as a motor or receiver. In brief, a selsyn device is mechanically similar to a miniature bipolar rotating-field, three-phase alternator. The rotor is wound with a single-phase concentrated winding, while the stator consists of a three-circuit distributed Y-connected winding. Electrically, the selsyn device acts as a transformer, and voltages and currents are all single-phase. The

Fig. 1—Above—Drilling units of automatic dynamic balancer are controlled directly by the signals to their controlling selsyns

Fig. 2—Right—Typical selsyn motors are similar in appearance to small fractional-horse-power motors



rotor winding is excited from an alternating-voltage source. By transformer action, voltages are induced in the three elements of the stator winding, the magnitude depending upon the physical angular position of the rotor. When the stator windings of the sender or generator are connected to that of the receiver or motor and the rotors are excited from a common a-c supply, the motor will assume the same physical angular position as the generator.

In a generator-motor system, the two selsyn rotors are excited from a common source. Assuming both rotors free to turn, they will take such a position that the voltages induced in the two stators are of balanced magnitude and displacement. Under this condition of stator-voltage balance between the two instruments, there is no circulating current in the stator windings. Should the rotor of one selsyn device be displaced by a certain angle and the rotor of the other is held in its original position, the stator-voltage balance is altered and a circulating current will flow in the windings. This circulating current reacting on the excitation flux provides a torque tending to turn the rotors of the machines to a position where the induced stator-voltages are again equal and opposite. Thus, with both rotors unrestrained, any motion given to the rotor of one selsyn device will be transmitted to and duplicated by the rotor of the second.

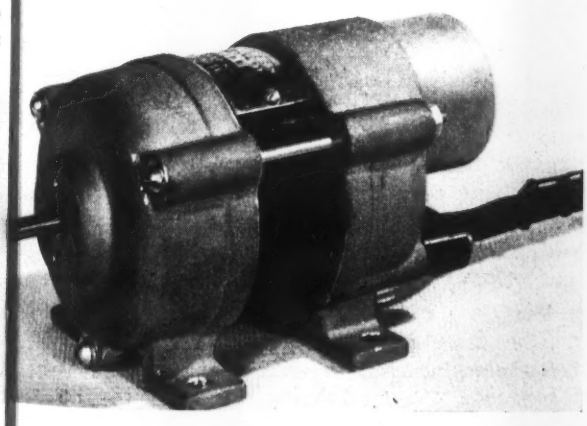
Selsyn Motors Require Mechanical Dampers

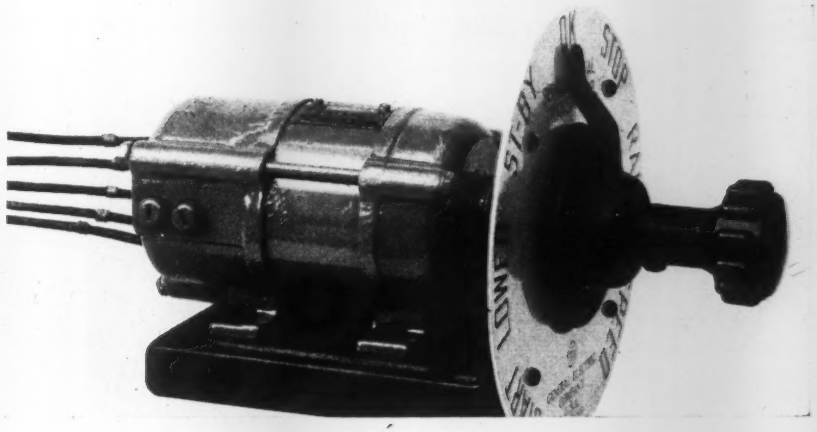
Before going too far with this description, it might be well to point out the difference between a generator and a motor. Since the generator is usually attached to a device having some inertia, there is little chance for small electrical differences to bring about instability which might result in mechanical oscillations or hunting. The receiver, usually used to operate instrument pointers, pens, or other low-inertia or low-friction loads, is free to move. For this reason, it is necessary to include a mechanical damper on the shaft of the selsyn motor. This provides rapid dampening as well as stability in operation. It is for this reason that motors can be used as generators, but generators cannot be used as motors.

Selsyns are available in a range of sizes from a few ounce inches of torque up to one hundred ounce-inches. In general, their appearance is similar to that of small fractional-horsepower motors, *Fig.* 2 The selsyns shown in *Fig.* 3 are typical of the type used in steel mills, on bridges, etc., to signal instructions to distant points.

By using selsyns to indicate the position of the stroke arm, large press manufacturers have been able to enclose the gears and belts in their latest machine designs. In *Fig.* 4, the selsyn transmitter can be seen mounted at the top of the press coupled to the stroke arm driving mechanism. The receiver is mounted at eye-level, convenient to the press operator. In previous designs the operator could visually observe the

Fig. 3—Below—Typical hand operated selsyn generator or transmitter unit





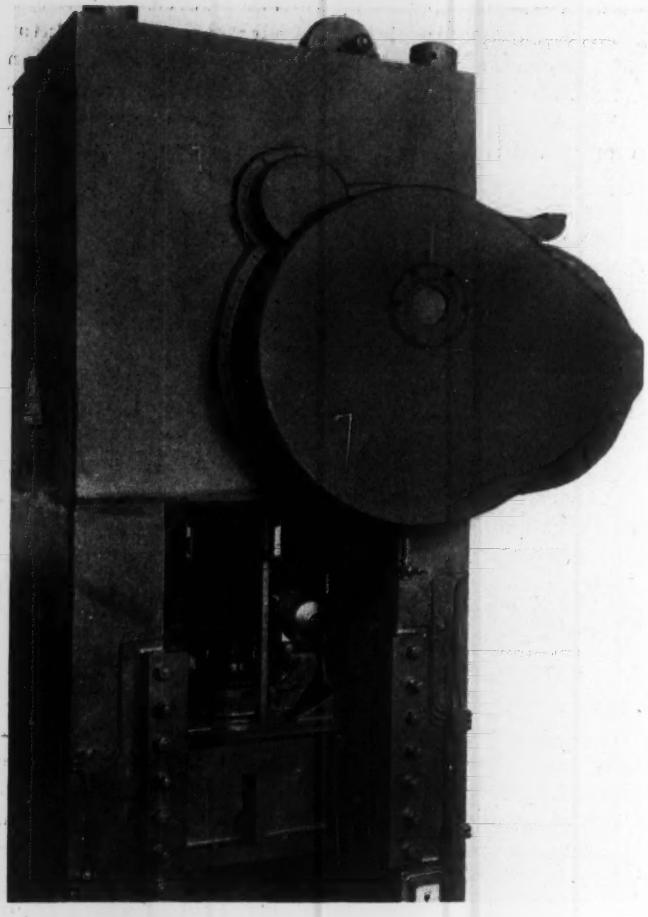


Fig. 4—Above — Large press utilizing transmitter at top and receiver at bottom right to indicate ram position

position of the stroke arm. However, selsyns provide an indicating means enabling designers to build less hazardous and better styled machines.

In the unusual machine shown in *Fig.* 5, selsyns transmit the unbalance of a motor crankshaft to drilling heads used to correct the unbalance. Placed in the dynamic balancer, the crankshaft is rotated at the desired speed and the six weighing dials, *Fig.* 6, coupled to six transmitters, are rotated to zero the balancing machine meter. These transmitters are electrically connected to selsyn motor receivers in the drilling machine and the receivers are coupled to cams which in turn control the depth of hole to be drilled in the crankshaft by each hydraulically operated drilling unit.

By the proper selection of a suitable transmitter, a large number of selsyn receivers in different locations may be operated from a single unit. Satisfactory operation over distances as great as five miles may be experienced at slightly reduced torque. The important factor under long distance operation is to keep secondary line resistance at a minimum. A No. 12 AWS wire will usually offer favorable operation over long distances. Another important factor to remember is that the primary excitation to the transmitter and receiver must have the same instantaneous phase relationship.

VOLTAGE REGULATOR: In textile testing machines, it is desirable to have a constant rate of loading applied to the fabric under test. One who has tried

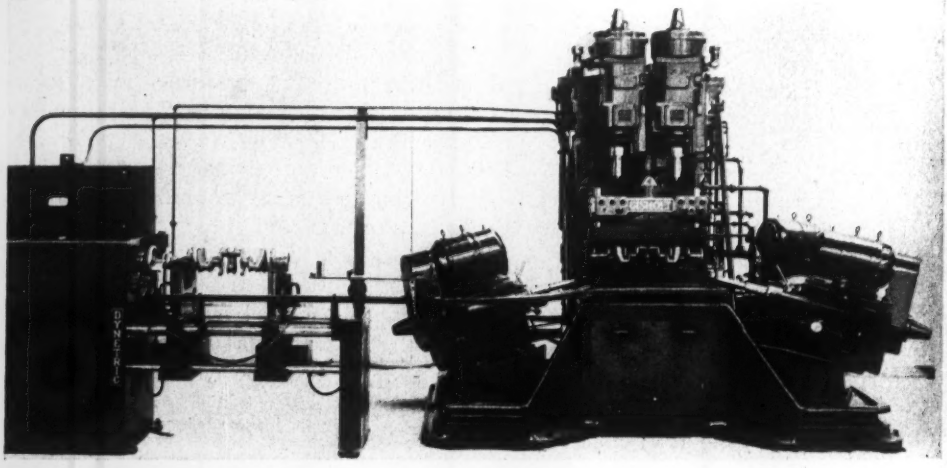
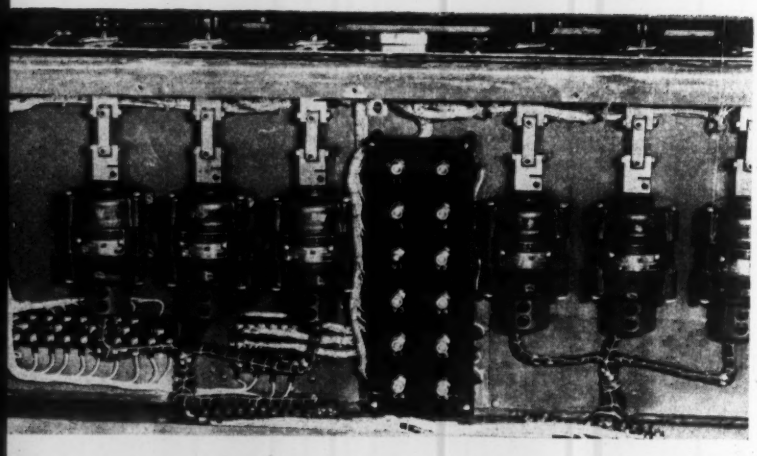


Fig. 5—Right—Selsyns in the dynamic balancer, left, transmit correct drilling depth information to the drills of this machine, see Fig. 1



to break a heavy string knows it is always easier to snap string rather than break it with a steady pull. It is to avoid this snapping action that a selsyn is associated with an electronic motor control to give a smooth even tension.

Most textile testing machines are of the pendulum type, and in this design the pull in the testing jaws is proportional to the sine of the angle through which the pendulum moves. By using a selsyn connected as an induction voltage regulator, it is possible to

Fig. 6—Left—Balancing machine transmitters which are adjusted in sequence to cause unbalance meter to read zero

obtain a voltage signal which follows a sine curve. By direct connection of the selsyn regulator to the torque balance shaft of the pendulum machine, it is possible to obtain a voltage signal proportional to the sine of the angle through which the pendulum is moved. By causing the pendulum to swing through its arc at a rate such that the sine of the angle increases in equal increments, it is possible to maintain accurate repetitive testing. To accomplish this, the output of the selsyn is connected to an electronic motor control which in turn acts to regulate the speed of the driving motor to give the desired constant tension.

DIFFERENTIAL: A selsyn differential system consists of a selsyn transmitter, a differential selsyn, and a selsyn receiver connected as shown in *Fig. 7*. Outwardly, the differential selsyn is similar to the other selsyns except that there are three connections to the primary winding instead of two. Three collector rings are provided to bring out the rotor-winding leads. Its physical form thus seems to be that of a miniature three-phase, wound-rotor induction motor. However, in normal operation its function is that of a single-phase transformer, and three-phase voltages and currents are not present. Its purpose is to modify the electrical angle transmitted by the selsyn transmitter so that the selsyn receiver will take up a position which will be either the sum or difference of the angles applied to the selsyn transmitter and the differential selsyn. Conversely, if two selsyn generators are connected through a differential selsyn and each is turned through an angle, the differential selsyn will indicate the sum or difference of the two angles.

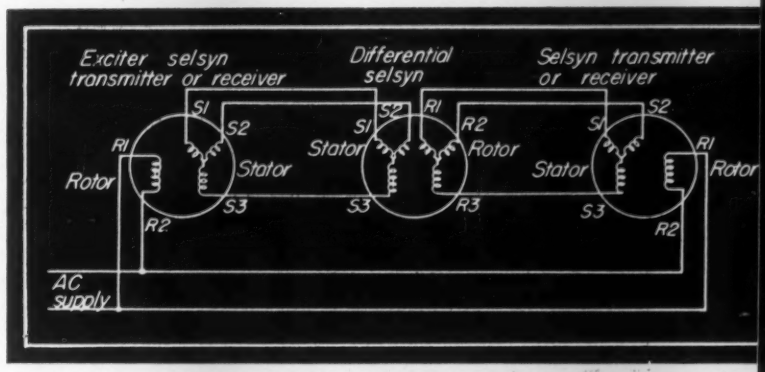


Fig. 7—Above—Electrical connection for a typical differential selsyn system

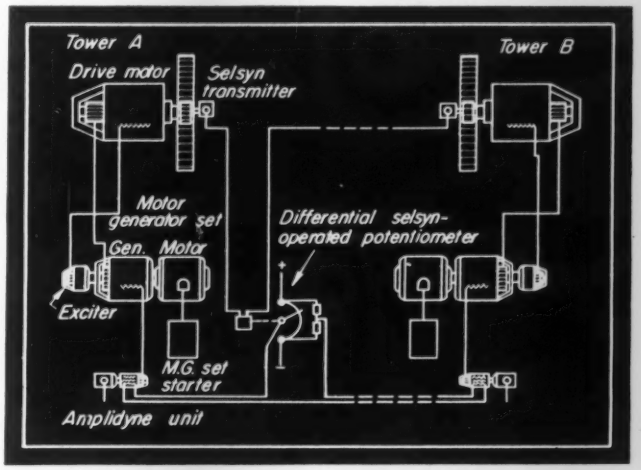


Fig. 8 — Above — Schematic connection diagram of lift-bridge drive using a differential system for maintaining alignment

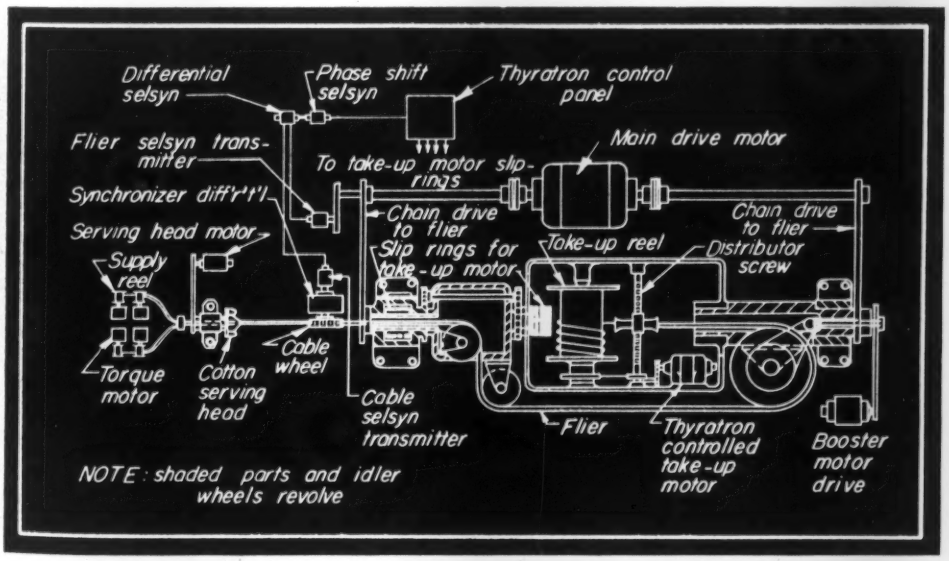


Fig. 9—Left—Schematic drawing of a wire stranding machine employing a differential system to maintain fixed proportional speed between the flier drive and cable take-up reel

Operation of a differential system is similar to that of the simple selsyn system. The voltage distribution in the primary winding is the same as that in the secondary winding of the exciter selsyn. If any one of the three selsyns is fixed in position and a second one displaced through a certain angle, the third, being free to rotate, will turn through the same angle. The direction of rotation can be reversed at will by reversing any pair of leads on either the rotor or the stator winding, respectively, of the differential selsyn. If any two of the selsyns are rotated simultaneously, the third will rotate through an angle equal to the algebraic sum of their movement, the algebraic sign being dependent not only on the physical direction of rotation of the rotors,

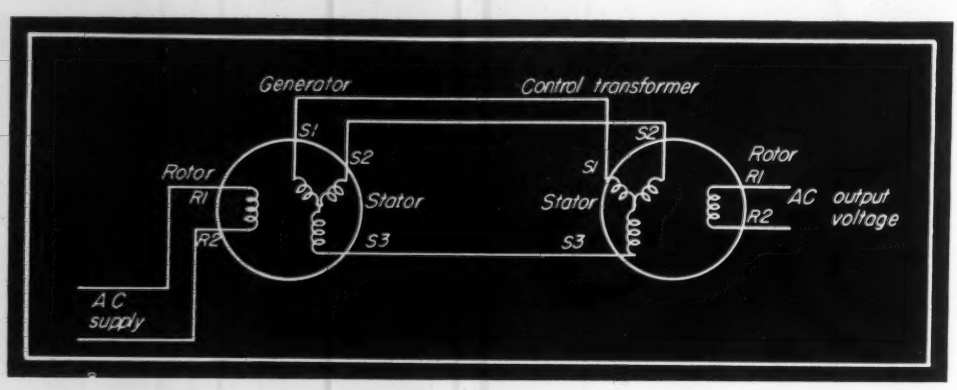
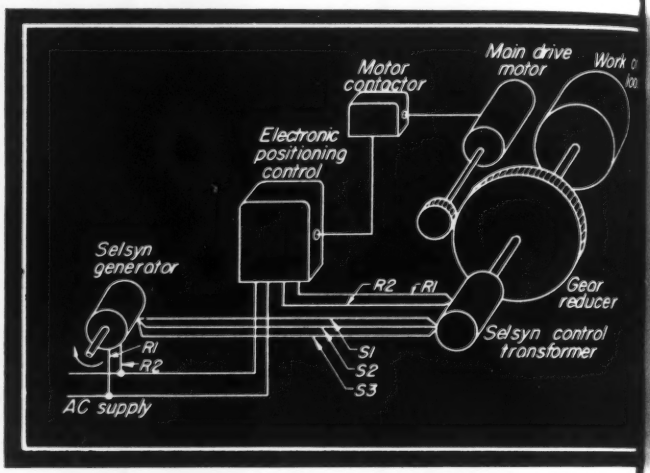


Fig. 10—Left—Connection diagram for typical selsyn generator and control transformer type system

Fig. 11—Below—Typical diagram for a positioning control system using a selsyn generator and a selsyn control transformer



but also on the phase relation or electrical connection of the winding.

Since the differential selsyn does not have a connection directly to the source of excitation, its excitation current must be furnished through one or both of the standard selsyns to which it is connected. The general practice is to supply this excitation to the primary winding only. It is obvious that since the selsyn connected to the differential stator supplies this excitation current, it must be designed to carry this extra load without overheating. A design of selsyn, commonly designated as a selsyn exciter, is used. It may function in the system either as a transmitter or as a receiver.

A selsyn differential system used in conjunction with amplidyne motor sets will be used on the new Passaic River vertical-lift bridge. Here, it will be necessary to keep the 2½ million pound, 220-foot span in close alignment. To do this, a selsyn transmitter will be coupled to each tower driving motor as shown in *Fig.* 8. These selsyns are then connected to a differential which is in turn called upon to operate a small potentiometer connected in the amplidyne field supply. Any difference in position between the driving motor in tower *A* and tower *B* will cause the differential and potentiometer to rotate. Rotation of this potentiometer is such that one amplidyne field will be strengthened and the other weakened to provide a correction that will hold the ends of the bridge to within 1½ inches of perfect alignment. The amplidynes are connected to control the output of the motor generator sets supplying power to each tower driving motor.

Electronic Control May be Desirable

In order to translate the output of a differential system into some form of electronic control, the potentiometer used in the lift bridge application may be replaced with the voltage regulator selsyn described in the textile testing machine. With this type of a differential system it is possible to measure the difference between two rotating shafts, translate this difference through a voltage regulator to an electronic motor control, which in turn causes a correction motor to operate and bring about a balance. Such a system might have been used in the lift bridge application, however, because of economics the amplidyne was preferred. In smaller motor applications the electronic method will be found more desirable.

In a telephone cable stranding machine it is important to have a constant pitch in the twisted pairs of wires. To accomplish this, a selsyn differential system is used to maintain a fixed proportional speed between the flier drive and the cable take up reel, *Fig.* 9. A selsyn transmitter is coupled to the cable wheel to give a signal of cable speed. A similar type transmitter is coupled to the flier or strander drive. The transmitters are connected to a differential which is in turn mechanically coupled to a selsyn voltage regulator. This regulator is connected in the grid phase-shift circuit of the electronic motor control, which panel supplies the power to the take-up reel driving motor. The equipment is adjusted to maintain a fixed proportion between the cable speed and the flier drive and any deviations in cable speed are corrected by the take-up motor.

In a selsyn system such as that used in the wire strander, it is advisable to employ mechanical stops at the limits of the phase-shift selsyn. Thus with the use of a slip clutch, it is possible to take care of the condition where the transmitters are far out of synchronism and the differential operates in continous rotation.

GENERATOR-CONTROL TRANSFORMER: Another basic type of selsyn system comprises a selsyn generator and a selsyn control transformer, *Fig.* 10. The control transformer used for voltage indication is mechanically similar to the selsyn generator. The rotor in the control transformer is not excited but acts as

a source of a-c voltage. It is for this reason that the term transformer has been given to this selsyn. The output or rotor voltage is sinusoidal and is used as a voltage signal to operate electronic devices.

Because a selsyn control transformer is designed to operate with a low volt-ampere burden, a minimum of power will be required of the selsyn generator. A well designed combination of generator and control transformer will have an error less than half a degree. By paralleling one system with another geared 16:1 to 36:1, it is possible to multiply the accuracy of a given system and obtain much smaller overall system errors.

Today, the combination of generator and control transformer selsyns is being applied to remotely operate valves, gates and dampers, and to position work in welding fixtures and machine tools. Typical interconnections between the selsyns, electronic controls, and rotating interconnections between the selsyns, electronic control and rotating equipment associated with this control is shown in *Fig.* 11. The selsyn generator is positioned mechanically or manually to that position to which the work is to be rotated or moved. Because the control transformer is mechanically connected to the work an "out of correspondence" signal is given to the electronic positioning control. The control in turn operates the forward or reverse contactor which energizes the driving motor bringing the work into a position corresponding to that of selsyn generator.

In a metal printing process it was possible to replace gears, chains, automatic clutches, and adjustable sprockets with a system employing selsyns. Because the printing process involved five different machines—feeder, printing press, varnish coater, baking oven, and a stacker—it was difficult to maintain register between all five operations with a suitable mechanical means. As a number of adjustments were required under the old method, the selsyn differential was called upon to supply an easily adjustable error or correction signal (*Fig.* 12). The output of the control transformer was coupled to an electronic amplifier controlling the field of an amplidyne generator, which in turn, supplied power to the driving motor. In *Fig.* 12 is also shown a line diagram of the process as well as a schematic diagram of the electrical connections. The press is used as the master machine to which the selsyn generator or exciter is coupled. This generator is then used to excite four control transformer selsyns coupled to the feeder, varnish coater, and baking oven drives.

Disregarding the differential selsyn for the moment, the generator and each control transformer, acting through the electronic amplifier and amplidyne, will operate in synchronism. Each time one of the drives attempts to speed up or fall behind, the control transformer signals for more or less speed to keep that drive in step with the master generator on the press.

By placing a manually-operated differential between each control transformer and the generator, it is possible to compensate for speed variations and changes in sheet size. If, for some reason, stopping the machine becomes necessary, it is possible to restart and synchronize to the next sheet by the simple adjustment of a knob connected to the differential shaft.

If all five machines were operated from a line shaft, special clutches would be required to operate any one machine or group of machines at one time. By simply feeding a dummy 60-cycle voltage into any one or a group of amplifiers, it is possible to operate each machine separately.

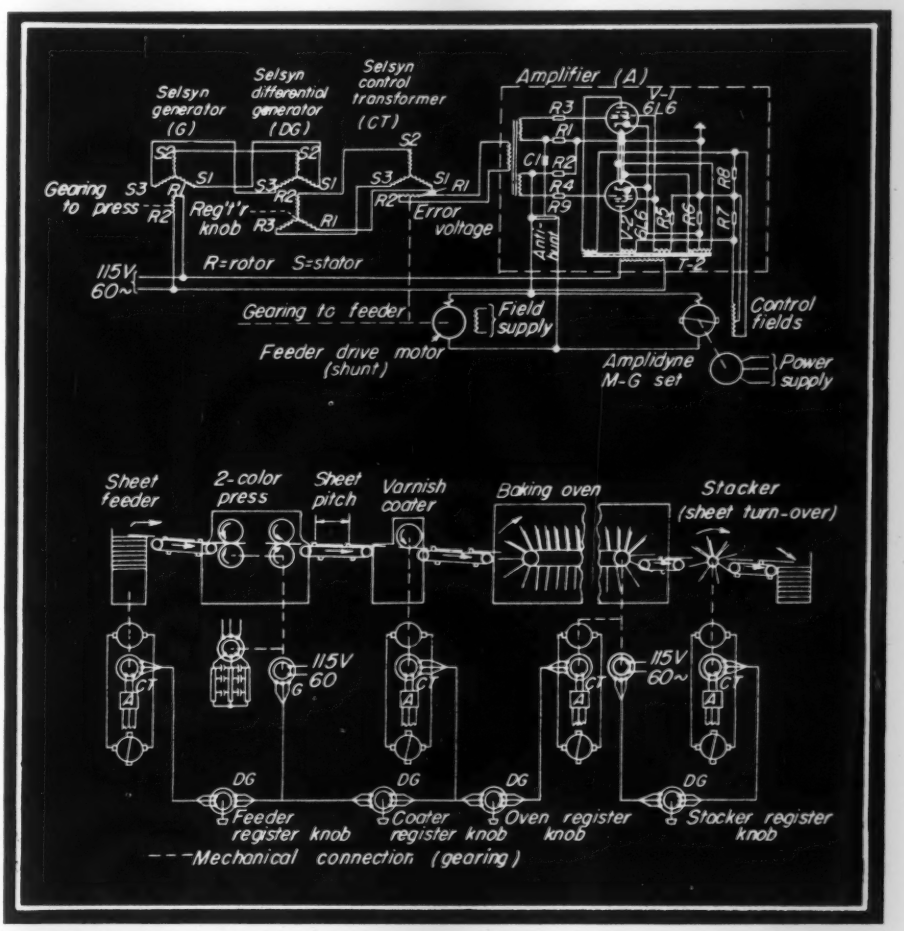
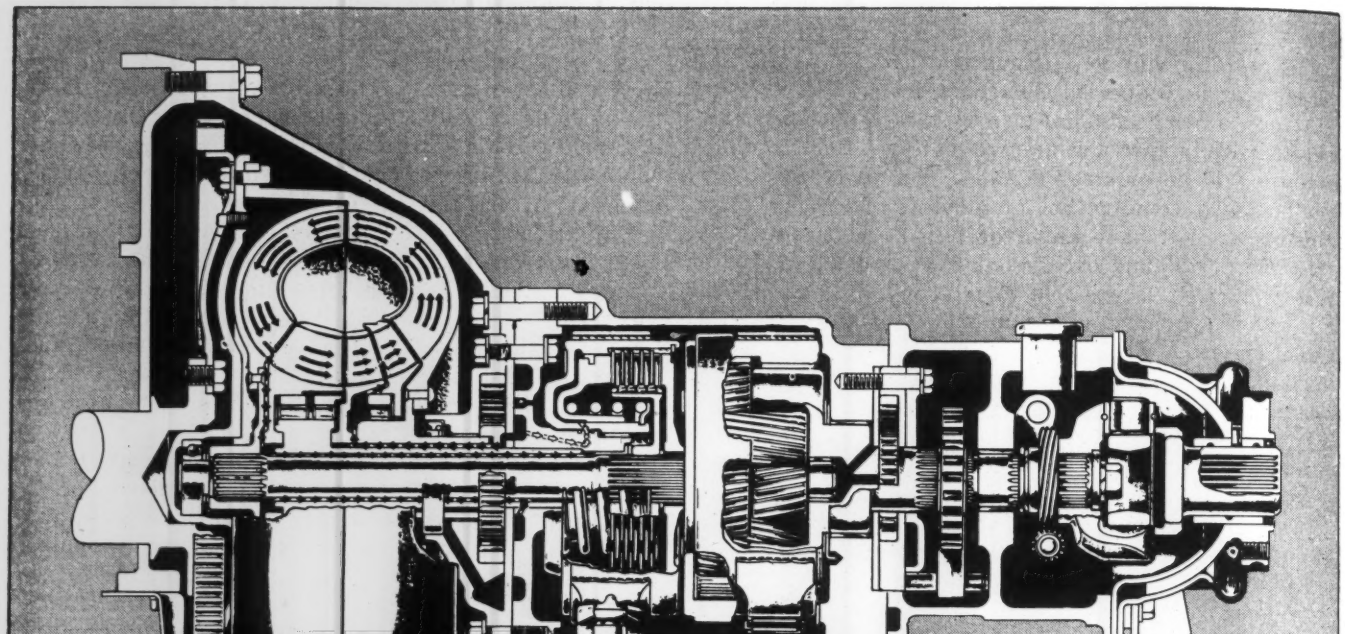


Fig. 12 — Right — Metal printing processing machine and selsyn generator - differential - control transformer system employed. Press is used as master machine



Torque Converter Drive

. . . for new Buicks provides infinitely variable speed ratios, supplemented by emergency low and reverse ratios furnished by planetary transmission

MATING of the internal combustion engine and the hydraulic torque converter to provide a drive of exceptional smoothness and flexibility has been so successful in the industrial field that ultimate adaptation to passenger cars has long seemed a foregone conclusion. First of the American passenger car makers to develop such a drive is Buick, which is now offering a torque converter drive called the Dynaflow, *Fig.* 1, as optional equipment on the 1948 Series 70 Roadmaster cars.

Fig. 1—Top—Torque converter at forward end provides all changes in speed ratio required for normal driving. Planetary gear unit gives the driver control over emergency low, neutral and reverse conditions

Characteristic of the torque converter is its inherent ability to provide torque multiplication as the ratio of output to input speeds decreases, *Fig.* 2. In a conventional transmission, or even in an automatic transmission like the Hydramatic, this is done in three or four finite steps which must be manually or automatically selected. The torque converter, on the other hand, provides an infinite series of ratios which are self-adjusting at all times according to the relative engine and wheel torques. The Dynaflow provides a torque multiplication from one up to at least 2.24, which is adequate for most driving conditions, including fast acceleration and hill climbing.

Supplementing the torque converter is a planetary

gear transmission under the control of the driver, who can select any of several alternatives— driving range, emergency low, neutral, or reverse. In the driving range the planetary unit provides a one-to-one ratio; in emergency low and reverse there is a reduction of 1.82 to one. When the torque converter is operating at its maximum ratio of 2.24 to one the overall torque multiplication in low and reverse is therefore 2.24 × 1.82 or about 4.1.

Torque Converter Requires no Controls

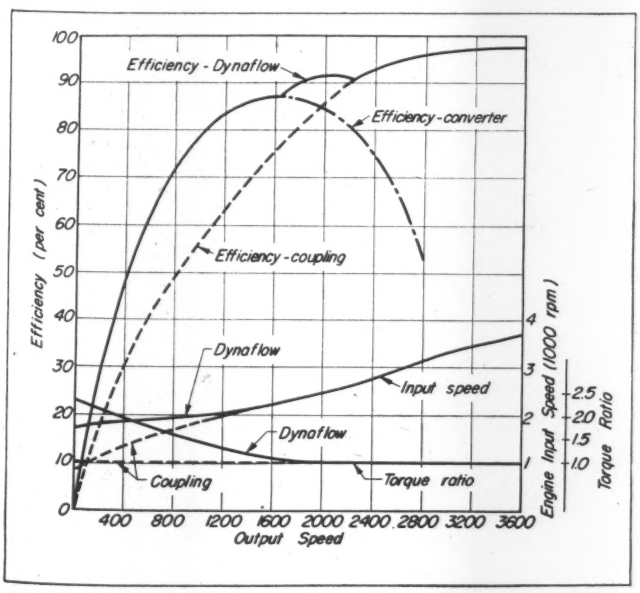
A longitudinal section through the complete Dynaflow transmission is shown in *Fig.* 1. The torque converter at the forward end is completely self-adjusting and requires no controls. Hydraulic fluid is circulated through it by either or both of two gear-type pumps. The planetary transmission incorporates two friction band brakes and one friction disk clutch, engagement or disengagement of which changes the speed ratio according to the will of the driver. Control brakes and clutches are actuated by hydraulic cylinder and piston mechanisms. The mechanical parking lock is operated by the same lever on the steering column which is used to select the desired forward, reverse, or neutral conditions.

A cross section through the torque converter unit is shown in *Fig.* 3, from which it may be observed that the unit consists of five elements—a turbine, two stator elements, and two pump elements. By contrast, a conventional fluid coupling to which it bears a superficial resemblance has only two elements —one turbine and one pump. The stator elements are necessary to absorb the torque reaction when operating as a converter, but are mounted on overrunning clutches which permit them to freewheel when the torque ratio drops to one. This occurs whenever the propeller shaft speed exceeds 85 per cent of the engine speed, and the converter then functions simply as a fluid coupling. Effect of this is illustrated in the performance chart, *Fig.* 2.

The cutaway view in *Fig.* 4 shows how the stator vanes are curved so as to re-direct the fluid flow from the turbine back to the pump. At high torque ratios fluid pressure on the backs of the blades locks the stator elements to the transmission case through the action of the freewheeling clutches, *Fig.* 3. As the speed difference becomes less, fluid pressure on the back of the secondary stator blades drops to zero, permitting the secondary stator to freewheel. As the turbine speed comes up to 85 per cent of the pump speed the fluid reaction forces on both stators drop to zero and both stators revolve at such a speed as to offer least resistance to fluid flow.

Somewhat similar action occurs in the pump. The secondary pump element is connected to the primary pump element through an overrunning clutch. At high torque ratios (large speed differences) the discharge velocity from the turbine through the stators is so high that it tends to push the secondary pump at a faster speed than the primary pump, which is

Fig. 2—Performance chart for Buick drive shows how ability to change from torque converter to simple coupling and vice versa increases the efficiency at all speeds



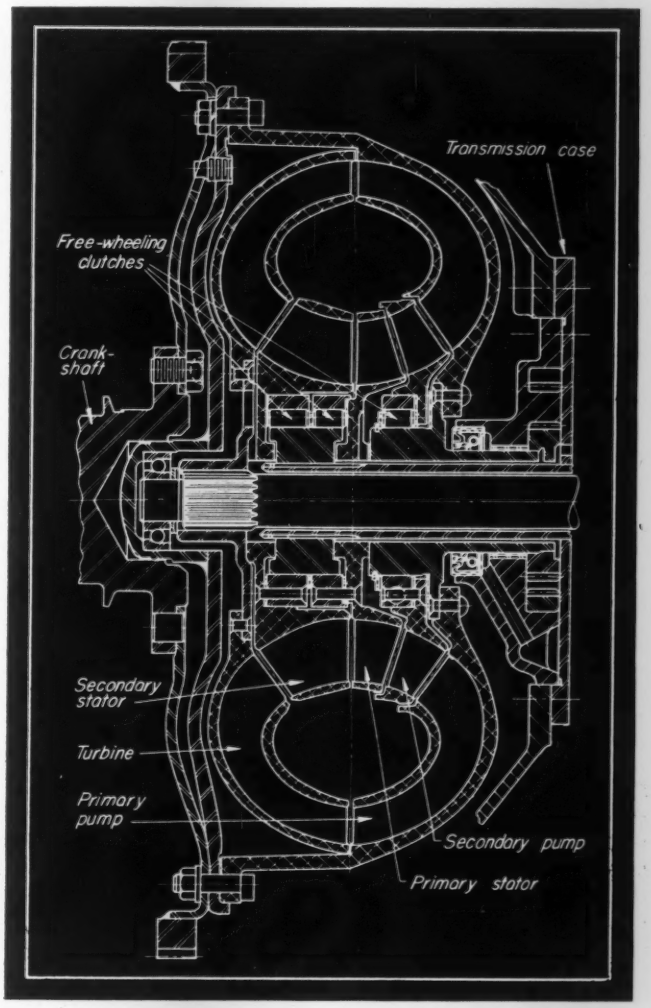


Fig. 3—Torque converter incorporates five elements. For torque conversion one or both of the stator elements lock to the case. Acting as coupling all five elements rotate and there is no torque conversion

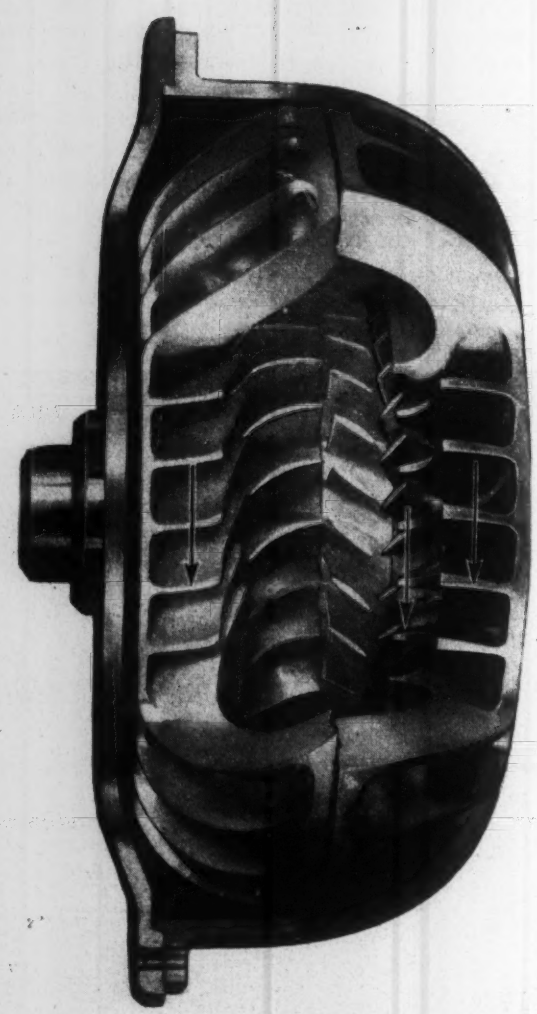


Fig. 4—Above—Cutaway view shows curvature of vanes in the various elements of the torque converter. Arrows show motion of pump and turbine elements when in the torque converter range

directly connected to the engine. The freewheeling feature permits the secondary pump to overrun the primary and thus, as it were, get out of the way of the fluid and permit higher efficiency. As input and output speeds become more nearly equal and the velocity of fluid flow decreases, the pressure on the back of the secondary pump disappears and overrunning stops, the secondary pump becoming locked to the primary. The transitions from one stage of operation to another occur smoothly and gradually. The five torque converter elements are aluminum plaster-mold castings.

The planetary transmission unit is shown in cross section in *Fig.* 5, while *Fig.* 6 shows part of the assembly, the ring gear being removed. There are two sun gears—the driving sun gear at the rear, which is splined to the input shaft, and the low-range reaction gear, which is connected to one element of a friction disk clutch. The other element of the clutch is splined to the input shaft, so that when the clutch is engaged the gear train becomes locked and the whole assembly rotates with the input shaft. The low-range reaction gear also is connected to a brake drum which can be locked to the housing when emergency low gear is required. The planet pinions are arranged in three pairs, each pair consisting of a low planet pinion, *Fig.* 5, engaging the driving sun gear, and a reverse planet pinion which engages (a) the low planet pinion, (b) the low-range reaction gear, and (c) the reverse gear. The low planet pinions are approximately twice as long as the other gears to permit engagement with both the driving sun gear and the reverse planet pinions.

From *Figs.* 5 and 6 the action of the planetary transmission may be followed. In emergency low gear the forward brake band is applied, locking the low-range reaction gear to the housing. In reverse the rear brake band is applied, locking the reverse

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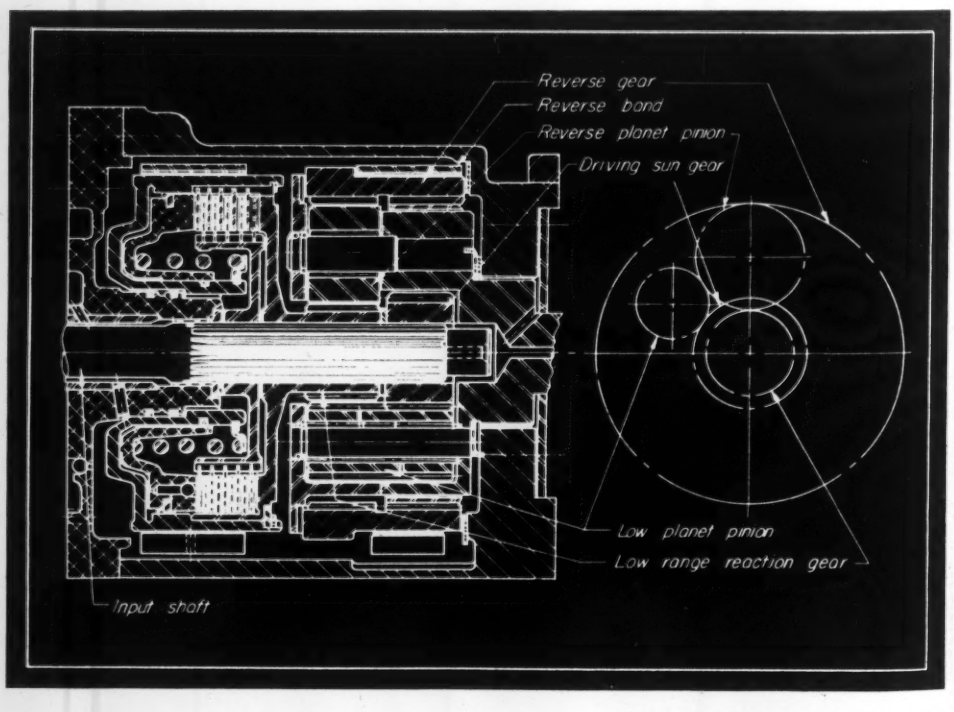

Fig. 5—Right—Cross section through planetary unit shows how low-range reaction gear can be locked to the case to provide emergency low or to the input shaft to provide 1:1 ratio. Reverse gear is locked to the case when reverse ratio is desired

Fig. 1—Shaft axis at an angle, tapered gear effects drive through mesh with conventional spur pinion

Designing Tapered Gears

Graphical construction and mathematical analysis are explained for both on-center and offset types

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A TAPERED gear may be defined as one which meshes with a conventional cylindrical spur or helical gear and whose axis, at an angle to that of the cylindrical gear, may or may not intersect the axis of the cylindrical gear. If the two axes are at right angles, the tapered gear becomes a face gear, either "offset" or "on center", depending on whether or not the axes intersect. Tapered gears may also be "on center" or "offset", the former having intersecing axes and the latter, nonintersecting axes. If the angle between the axes is allowed to diminish until the axes are parallel, the tapered gear becomes an ordinary cylindrical gear. This article will be confined to tapered gears meshing with a conventional involute spur gear, *Fig.* 1.

To make clear the action between a tapered gear and pinion, the same technique will be used as was developed in a recent article on face gears (MACHINE DESIGN, April, 1947). In place of gears, the schematic drawings show pairs of friction rolls of very narrow width having diameters equal to the pitch diameters of the tapered gear and pinion. The action of an on-center tapered gear and pinion, shown diagrammatically by such friction rolls in *Fig.* 2, will be considered first.

Let N_p and N_t equal the numbers of teeth in the pinion and tapered gears, respectively, ω_p and ω_t equal

ratio such that,

$$K = \frac{N_p}{N_t} = \frac{\omega_t}{\omega_p} \quad \ldots\ldots (1)$$

If r_1 and R_1 equal the active pitch radii of the pinion and tapered gear, the corresponding tangential velocity is $r_1\ \omega_p$ and $R_1\ \omega_t$. These tangential velocities may be represented by lines drawn perpendicular to the radii at the common tangent. Since $r_1\ \omega_p = R_1\ \omega_t$ it follows that

$$\frac{r_1}{R_1} = \frac{\omega_t}{\omega_p} = K \quad \ldots\ldots (2)$$

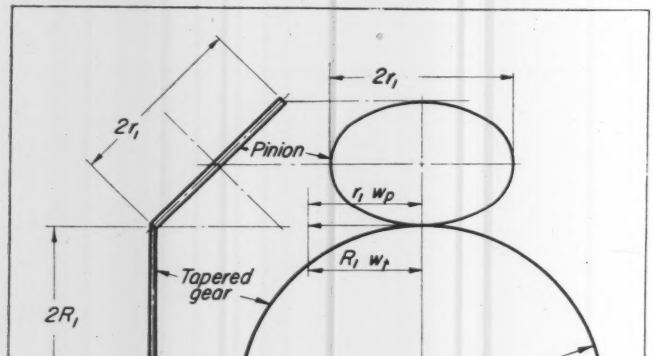


Fig. 2—Friction rolls show diagrammatically the action of on-center tapered gear and pinion

By establishing a number of these narrow rings, it is possible to define the pitch surfaces of the gears. The diameters of the rings must all satisfy Equation 2. From *Fig.* 3 it will be seen that

$$\sin\phi = \frac{r_1 + R_1\cos\phi}{C} = \frac{KR_1 + R_1\cos\phi}{C}$$

$$= \frac{KR_1}{C} + \tan V\cos\phi = \tan V\ (K + \cos\phi)$$

$$\tan V = \frac{\sin\phi}{K + \cos\phi} \quad \ldots\ldots (3)$$

By inspection, it can be seen that

$$K = \frac{\sin U}{\sin V} \quad \ldots\ldots (4)$$

$$\phi = U\ V \quad \ldots\ldots (5)$$

Since all contacts occur on line *OP*, this line, if allowed to revolve about the axis of each gear, will define the pitch surfaces of the tapered gear and pinion. The pitch surface of the tapered gear is a cone with apex at *O* and cone angle *V*, and the pitch surface of the pinion is a cone with apex also at *O* and cone angle *U*. The shaft angle equals the sum of the cone angles. Thus, by establishing any pitch radius on either gear, it is possible to find the corresponding pitch radius of the other.

Since the base radius of the pinion is constant for its entire face width, and since the pitch radius varies along the face width, it follows that the active pressure angle of the tapered gear and pinion must increase as the pitch point moves out from the cone center along the pitch cone. This condition is similar, although not so marked as, the one found in face gears. If B_i is any active pressure angle, a the base radius of the pinion, and r_i any active pitch radius of the pinion, then

$$\cos B_i = \frac{a}{r_i} \quad \ldots\ldots (6)$$

To find the shape of the tapered gear tooth at any point R_1 a section is taken perpendicular to the axis of the pinion. In this plane the new active pitch radius of the tapered gear, called R_c, is found thus

$$R_c = R\sec\phi \quad \ldots\ldots (7)$$

The base radius of the tapered gear, a_t, becomes

$$a_t = R\sec\phi\cos B$$

$$a_t = \frac{a}{K}\sec\phi \quad \ldots\ldots (8)$$

It will be seen that the base radius of the tapered gear, as well as the pinion, is constant.

A pinion with a tooth space on the centerline is shown in *Fig.* 4. The pitch radii, r_1 and r_2, at each end of the face width are found from Equation 2. Tangents are drawn from these points to the base radius a, these tangents forming the active pressure angles B_1 and B_2. The corresponding radii R_{c1} and R_{c2}, determined from Equation 7, are then laid off along the centerline as shown and the radius a_t is drawn from each center. Involutes are then drawn from the base radii tangent to the involutes of the pinion, contact taking place along each line of action. These involutes represent the developed shapes of the tapered gear at R_1 and R_2, in sections normal to the pinion axis.

Except for special cases, the working depth between a tapered gear and pinion should be constant for the entire face width. The outside radius of the tapered gear, in a plane normal to the pinion axis can, therefore, be determined by layout in *Fig.* 4. If l_o is the outside radius in this plane, l is the actual outside radius, and N is the outside radius of the pinion minus the working depth, then

$$l_{o1} = (r_1 + R_{o1}) - N$$

$$l_{o1} = l_1\sec\phi$$

$$l_1 = (r_1 + R_{o1} - N)\cos\phi \quad \ldots\ldots (9)$$

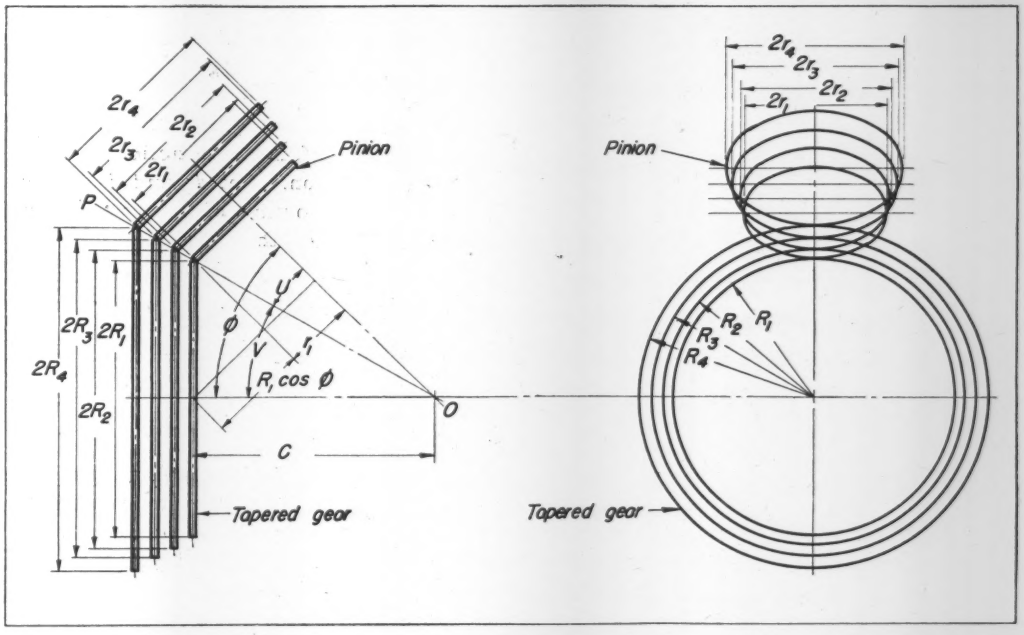


Fig. 3—Pitch surfaces of on-center tapered gear and pinion are defined by line OP along which all contacts occur

The outside cone angle, for dimension purposes, is ϕ.

In general, it is not considered advisable to use a tapered gear that has pointed teeth. Therefore, it is important to establish the diameter at which the tapered gear tooth becomes pointed. This can be done by trial and error, thus establishing a pitch radius R_t, and a corresponding outside radius l_t, at which the tooth becomes pointed. R_t and l_t may be considered the maximum desirable pitch radius and outside radius of the tapered gear. Likewise, by trial and error, minimum pitch and outside radii may be established as those radii at which involute interference takes place. If the minimum pitch radius is R_1, the maximum desirable face width of the tapered gear, measured along its axis, is

$$F_{max} = \frac{R_t - R_1}{\tan V} \qquad (10)$$

However, as in the case of face gears, it is sometimes impossible to use a tapered gear with radii established in this manner. Due to the fact that the tapered gear must be generated by a pinion-shaped cutter, reciprocated on its axis, having the same number of teeth (or very nearly the same) as the mating pinion, there is a possibility that portions of the tapered gear tooth will be trimmed away due to the generating action. To determine whether such trimming will occur, a diagram such as that of *Fig.* 5 is developed, on which

$$l_a = (l_c \text{ at small end}) \sec^2 \phi$$

$$g = (\text{working depth}) \sin \phi$$

The intersection of l_a and the pinion O.D. is projected parallel to the line of centers until it intersects the line represented by α_t, thus determining R_{ci}. The

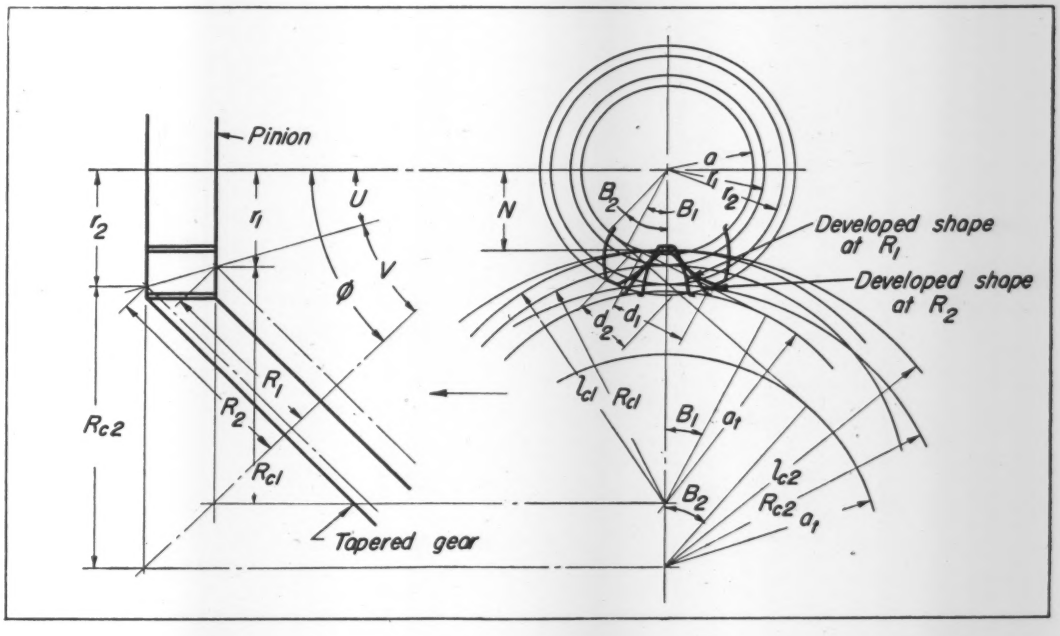


Fig. 4—How shapes of teeth at front and rear faces of on-center tapered gear are developed

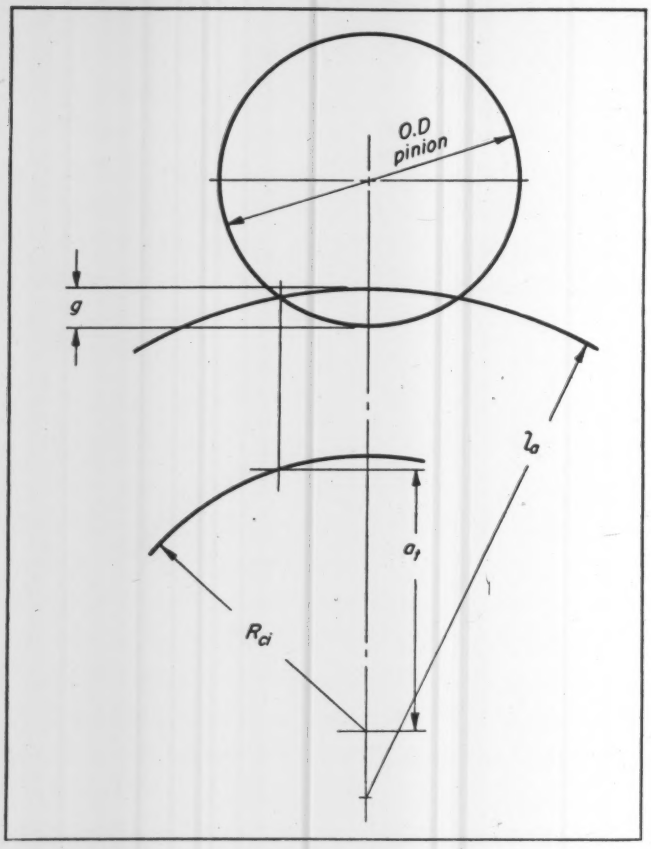


Fig. 5—Above—Determining if portions of on-center tapered gear teeth will be trimmed away when generated

Fig. 6—Below—Offset tapered gears have tangential velocities that do not coincide and are unequal

pitch radius at which trimming will start is $R_{ci} \cos \phi$. Except for large values of ϕ, trimming will be found to be negligible compared to natural involute interference.

From *Fig.* 4, it is possible to determine the amount of involute action between the tapered gear and pinion. Let d_1 and d_2 represent the length of the active portion of the line of action at each end of the face width. If m_1 and m_2 represent the contact ratio, then

$$m_1 = \frac{N_p d_1}{2\pi a} \qquad (11)$$

$$m_2 = \frac{N_p d_2}{2\pi a}$$

The foregoing analysis has been made on an external type on-center tapered gear. It readily will be seen that the same principles apply to an internal type tapered gear. Equation 3 becomes

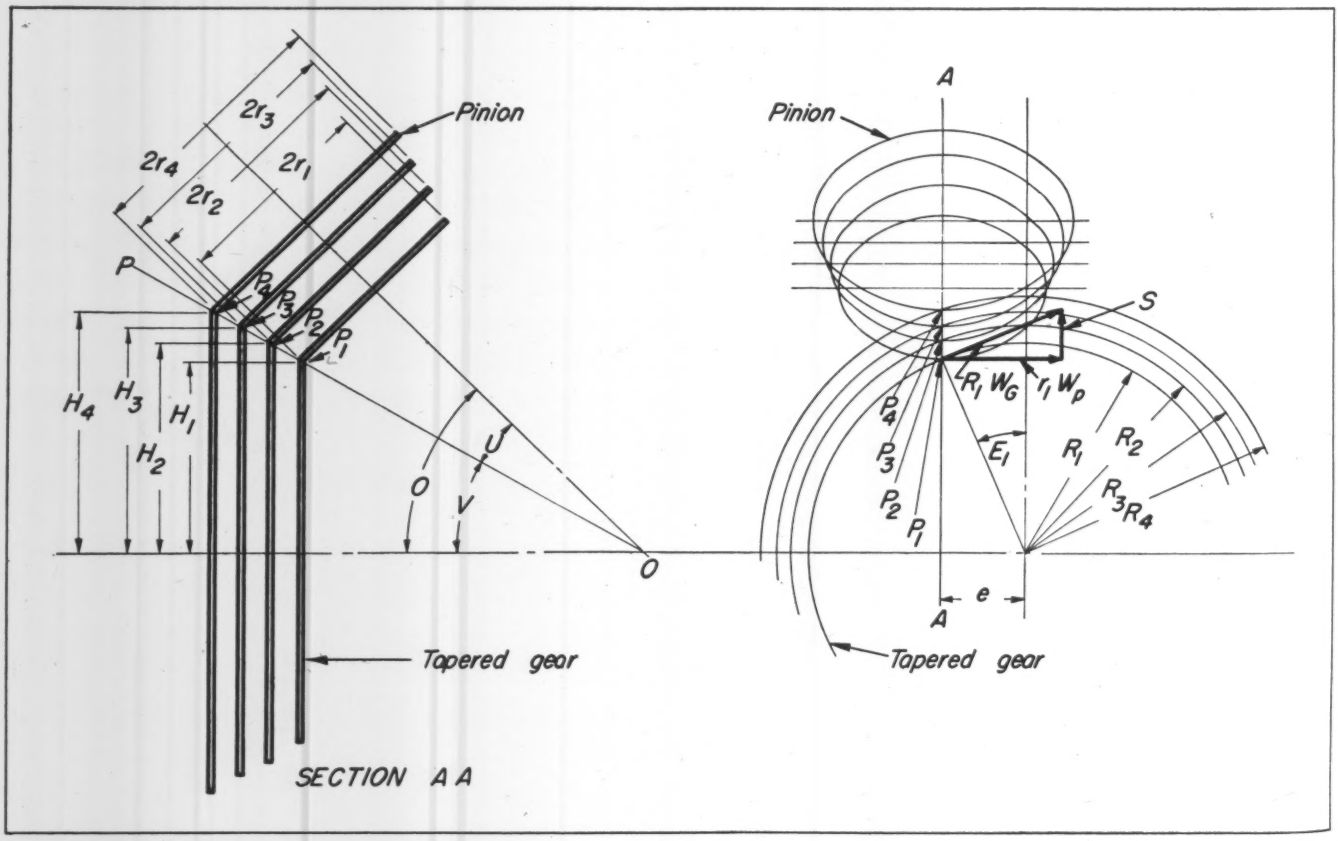
$$\tan V = \frac{\sin \phi}{\cos \phi - K} \qquad (3_1)$$

Equation 4 remains the same and Equation 5 becomes

$$\phi = V - U \qquad (5_1)$$

In all other respects, the same formulas apply to both external and internal type tapered gears.

The designer should always try to keep the involute action more than 1.00 for as much of the face width as possible. This should not be difficult for low shaft angles and comparatively narrow face widths. How-



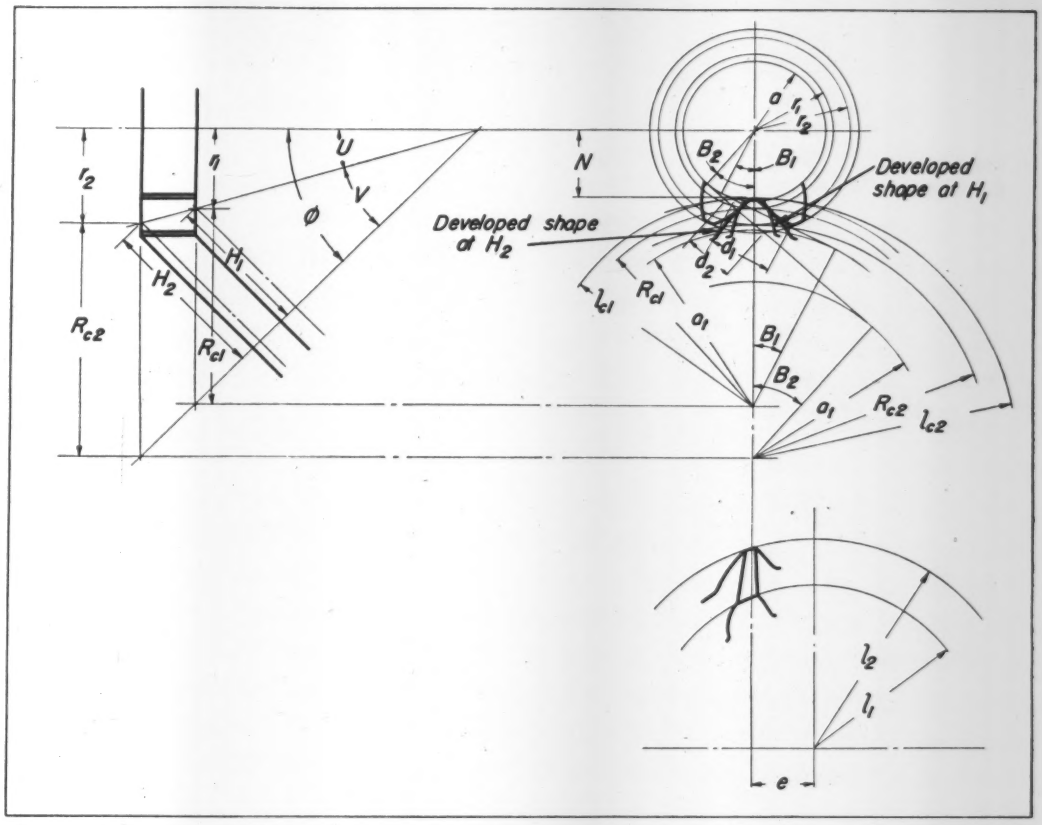


Fig. 7 — Developed shape of offset tapered gear tooth from front to rear faces

ever, even when it is necessary to use wide face widths and higher shaft angles, it will be found advisable to use that portion of the gear which provides for more than 1.00 contact ratio over more than half the face width.

So far the only tapered gears and pinions considered have been those with intersecting axes. In considering the offset tapered gears and pinions (those with nonintersecting axes) the method of approach is similar to that used in the offset face gear, which may be considered as a special case of the general offset tapered gear. Again, two narrow friction disks are substituted for the gears as shown in *Fig.* 6. It will be seen that the directions of the tangential velocities, $R_1\ \omega_G$ and $r_1\ \omega_p$, no longer coincide and are now unequal. The relationship is

$$r_1\omega_p = R_1\omega_G \cos E_1$$

The sliding S, parallel to the pinion axis, is

$$S = R_1\omega_G \sin E_1$$

Substituting $K = W_G/W_p$ results in

$$r_1 = KR_1 \cos E_1 \quad \ldots\ldots (12)$$

Determine various values of H as shown in *Fig.* 6.

$$H_1 = R_1 \cos E_1 = \sqrt{R_1^2 - e^2}$$

$$r_1 = KH_1 = K\sqrt{R_1^2 - e^2} \quad \ldots\ldots (13)$$

By substituting a series of rolls, various contact points, P_i, are determined, and all the radii r_i and H_i must satisfy Equation 13. It will be seen that Equations 3, 4 and 5 are applicable if the offset pinion axis is taken as the reference line. It is now possible to determine the pitch radius of the pinion for any given pitch radius of the tapered gear. It will be seen that the pitch surface of the pinion (generated by line *OP* as it revolves about the pinion axis) is a cone, while the pitch surface of the tapered gear (generated by line *OP* as it revolves about the gear axis) is a hyperboloid of revolution.

To find the shape of the offset tapered gear tooth at any pitch radius, R_1, it is necessary to find the corresponding pitch radius, r_1, of the pinion, from which the active pressure angle, B_1, is found from Equation 6. A section is taken, as before, perpendicular to the axis of the pinion, and the active pitch radius of the tapered gear in this plane, R_{c1}, is found thus

$$R_{c1} = H_1 \sec \phi \quad \ldots\ldots (14)$$

a_t is found from Equation 8. *Fig.* 7 shows the shapes of an offset tapered gear tooth at the large and small ends of the tapered gear, meshing with a spur pinion, with a space of the pinion tooth on the center line. The method of construction is the same as before, if H is substituted for R in the on-center tapered gear.

Again maintaining a constant working depth throughout the face width, the outside radii in this plane are determined. Let l_{c1} be the outside radius in this plane, and l_1 the actual outside radius taken from the center of the gear, then

$$l_1 = \sqrt{(l_{c1} \cos \phi)^2 + e^2} \quad \ldots\ldots (15)$$

By trial, the diameters at which the tapered gear becomes pointed, and at which involute interference starts, can be determined. Involute action is found from Equation 10.

In dimensioning an offset tapered gear, it must be remembered that the angles U and V are taken about the offset pinion axis and not around the center of the gear. The actual cone angle, ϕ_a, for machining the tapered gear must be found. The face width, F, of the gear is

$$F = \frac{H_2 - H_1}{\tan V} \qquad (16)$$

The actual cone is

$$\tan \phi_a = \frac{l_2 - l_1}{F} \qquad (17)$$

or, by substituting from Equations 16 and 3

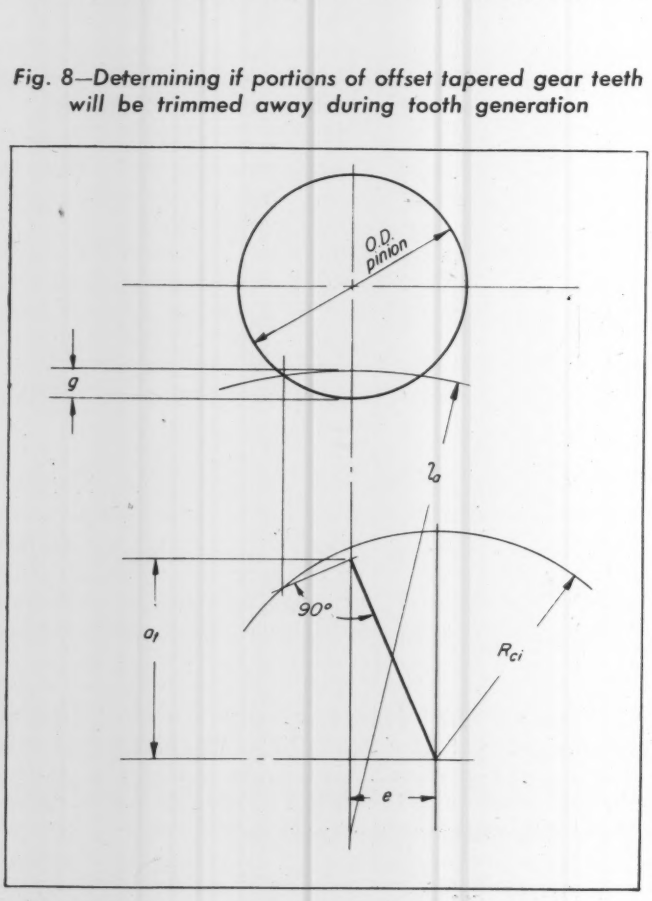
$$\tan \phi_a = \frac{(l_2 - l_1) \sin \phi}{(H_2 - H_1)(K + \cos \phi)}$$

The diameter on the tapered gear at which trimming occurs, due to the action of the reciprocating pinion-shaped cutter, can be found from *Fig.* 8. Here again

$$g = (\text{working depth}) \sin \phi$$
$$l_a = (l_c \text{ at small end}) \sec^2 \phi$$

The pitch diameter R_i at which trimming starts is

Fig. 8—Determining if portions of offset tapered gear teeth will be trimmed away during tooth generation



$$R_i = \sqrt{(H_{c1} \cos \phi)^2 + e^2}$$

The amount of involute action can be determined from *Fig.* 7, using Equation 11.

The designer, in choosing diameters and face widths of offset tapered gears, should bear in mind the same conditions, as to amount of involute action, mentioned previously in connection with on-center tapered gears. That portion of the face width which admits of the best working condition should be used.

There has recently been developed, at the Fellows Gear Shaper company, a method of cutting the tapered gear so as to produce a localized bearing, or crowned tooth, when operated with a spur pinion. This process enables the designer to allow for small shaft misalignments, slight deviations from theoretical cone distance, and small deflections under load. The crowning is incorporated entirely in the tapered gear, permitting the manufacturer to use established, conventional means for producing the spur pinion.

Small Turbine Generators Standardized

A RECENTLY redesigned family of Westinghouse high-speed, two-pole industrial turbine generators is now available in ratings ranging from 1000 to 7500 kw inclusive, at 480 to 13,800 volts. These units are standardized, the intent being to eliminate, as far as possible the need for building each unit as a special, "tailor-made" machine, and to promote off-the-shelf purchasing with its attendant advantages to both builder and user.

Standardization makes possible the production of better generators without increasing overall cost. Each variation from the standard design, no matter how trivial, usually entails additional engineering and construction work, which can increase the cost of the unit considerably. With design standardized, many improvements were made in production methods and, where applicable, simplified tabulated drawings were made for the ratings selected. Manufacturing is facilitated by the use of jigs and fixtures. For example, the component parts of the frame are premachined and accurately welded together in an assembly fixture.

Appearance has been modernized and styling is identical in all sizes; length of the units is reduced; accessibility to the collector and commutator is improved; and the exciter and collector are enclosed in a clean-lined housing. Various accessories, such as an electric indicating tachometer, an exciter air filter, and clamp-type solderless terminal lugs, are included as standard features.

Several special new features associated with the exciter are particularly effective in reducing installation costs and facilitating maintenance. The need for exciter ducts through the foundation has been eliminated by ventilating the unit from the generator through ducts incorporated in the exciter bedplate and generator end bell. In addition, filters clean the air before it is returned from the exciter to the generator.

Ring Deflection

. . . due to loads normal to the plane of curvature is calculated readily with the aid of curves presented in this article

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ANALYSIS of circular rings loaded normal to the plane of curvature, *Fig.* 1, has received relatively slight attention from engineers. This should not serve as a measure of the importance of the subject, however, since many circular elements so loaded are employed as essential components in machines such as helicopter control mechanisms for which the analysis which follows was developed.

The deflection equations presented in this article have been derived by the use of Castigliano's first theorem. It is assumed that the flexibility of the ring is such that the strain energy due to direct shear is small compared with that due to bending and torsion.

DEFLECTION OF A CONCENTRATED CENTRAL LOAD: The ring shown in *Fig.* 1 is hinged at A_1 and A_2, the line joining these points being a diameter. The concentrated loads P acting normal to the plane of curvature are applied at B_1 and B_2, the line joining these points being a diameter. Lines A_1A_2 and B_1B_2 are mutually perpendicular.

From the symmetry of loading, and the nature of the supports, it is evident that the reactive moments at A_1 and A_2 are equal. Applying Castigliano's first theorem (displacement at and in the direction of the load is equal to the first derivative of the strain energy with respect to the load), the deflection of the point of application of P along its line of action is given by the equation:

$$\Delta = \frac{\partial U}{\partial P} \qquad (1)$$

Since the loading is symmetrical, the strain energy for only one quadrant of the ring need be considered. The expression for the strain energy is:

$$U = 2\int_0^{\pi/2} \left(\frac{M_x^2}{2EI_x} + \frac{M_z^2}{2GK} \right) R d\theta$$

Substituting this in Equation 1,

$$\Delta = \frac{\partial U}{\partial P} = 2\left\{ \int_0^{\pi/2} \frac{M_x}{EI_x} \frac{\partial M_x}{\partial P} R d\theta + \int_0^{\pi/2} \frac{M_z}{GK} \frac{\partial M_z}{\partial P} R d\theta \right\} \qquad (2)$$

Cutting the ring at A_1, and restoring the reactive elements operating there before the cut was effected, namely, a vertical reaction $P/2$ and a moment $PR/2$, the moments M_x and M_z at any section D at an angular distance θ from OA_1 are:

Fig. 1—Circular ring hinged at A_1 and A_2 carries concentrated loads P on diameter at right angles to A_1A_2

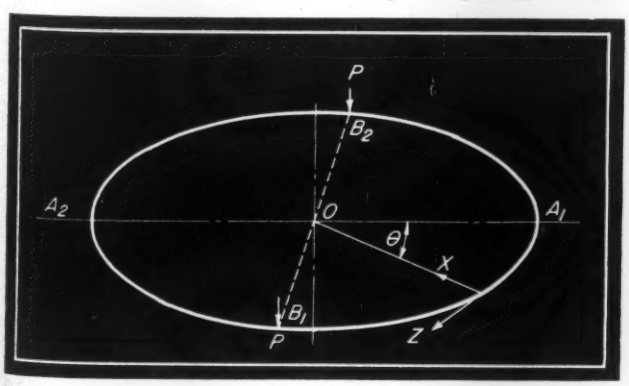
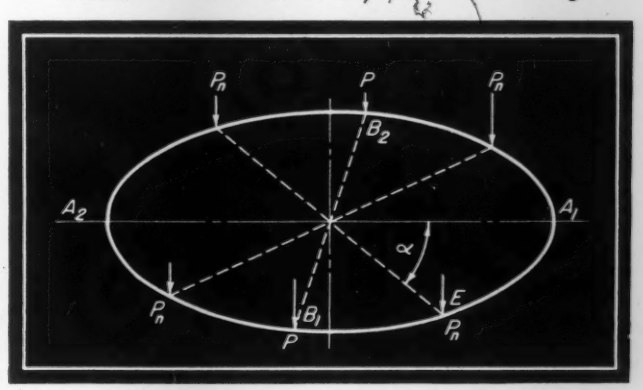


Fig. 2—Application of hypothetical loads P_n facilitates calculation of deflection at any point on the ring



$$M_x = \frac{P}{2} R \cos\theta - \frac{P}{2} R \sin\theta$$

$$M_z = \frac{P}{2} R \sin\theta - \frac{P}{2} R (1 - \cos\theta)$$

from which:

$$\frac{\partial M_x}{\partial P} = \frac{R}{2} (\cos\theta - \sin\theta)$$

$$\frac{\partial M_z}{\partial P} = \frac{R}{2} (\sin\theta + \cos\theta - 1)$$

Substituting these in Equation 2,

$$\Delta = \frac{\partial U}{\partial P} = \frac{PR^3}{2EI_x} \int_0^{\pi/2} (\cos\theta - \sin\theta)^2 d\theta + \frac{PR^3}{2GK} \int_0^{\pi/2} (\sin\theta + \cos\theta - 1)^2 d\theta$$

$$= \frac{PR^3}{2} \left[\frac{1}{EI_x} \left(\frac{\pi}{2} - 1\right) + \frac{1}{GK} (\pi - 3) \right] \quad (3)$$

If the cross-section is circular, $K = J = 2I_x$. Also taking $E = 2.6G$,

$$\Delta = \frac{PR^3}{2EI_x} \left[\left(\frac{\pi}{2} - 1\right) + 1.3 (\pi - 3) \right]$$

$$= 0.376 \frac{PR^3}{EI_x} \quad (4)$$

DEFLECTION OF ANY POINT on the ring due to the action of the central load P can be found by considering the deflection of any one of the concentrated hypothetical loads P_n shown in *Fig.* 2. The number of these loads and their location on the ring must be such as not to alter the symmetry of loading. The procedure adopted in the previous case is repeated and leads to a general deflection equation containing P_n; by equating P_n to zero a final equation for the deflection of any point is obtained. Cutting the ring at A_1 and restoring the reactive elements, namely, a vertical reactive force $(P/2) + P_n$ and a moment $M_{z_0} = (PR/2) + (P_n R \sin\alpha)$, the moments M_x and M_z at any section between A_1 and E are:

Nomenclature

- U = Strain energy
- M_x = Moment about X-axis
- M_z = Moment about Z-axis
- P = Concentrated load
- P_n = Hypothetical concentrated load
- Δ = Deflection
- θ = Angular distance of any section from OA_1
- α = Angular distance from OA_1 of any section
- I = Second moment of area about X-axis
- K = Factor dependent on the form and dimensions of the cross-section = Polar moment of inertia, J, for a circular cross-section
- E = Young's modulus of elasticity
- G = Modulus of rigidity
- ε = Offset from the center of twist at B_1 and B_2
- ε_1 = Offset from the center of twist at A_1 and A_2
- w = Uniformly distributed load
- R = Radius of ring

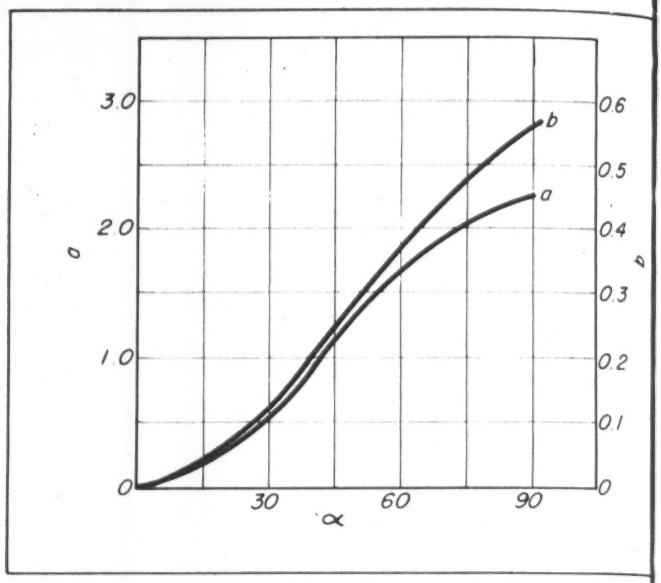


Fig. 3—Curves give constants for Equation 8, aiding calculation of deflection at any point on ring, Fig. 2

$$M_x = \left(\frac{PR}{2} + P_n R \sin\alpha\right) \cos\theta - \left(\frac{P}{2} + P_n\right) R \sin\theta$$

$$M_z = \left(\frac{PR}{2} + P_n R \sin\alpha\right) \sin\theta - \left(\frac{P}{2} + P_n\right) (1 - \cos\theta) R$$

For the section between A_1 and E,

$$\left|\frac{\partial U}{\partial P_n}\right|_{A_1 E} = \frac{1}{EI_x} \int_0^{\alpha} M_x \frac{\partial M_x}{\partial P_n} R d\theta + \frac{1}{GK} \int_0^{\alpha} M_z \frac{\partial M_z}{\partial P_n} R d\theta \quad (5)$$

Considering section EB_1, the moments M_x and M_z at any section between E and B_1 are:

$$M_x = \left(\frac{PR}{2} + P_n R \sin\alpha\right) \cos\theta - \left(\frac{P}{2} + P_n\right) R \sin\theta + P_n R \sin(\theta - \alpha)$$

$$M_z = \left(\frac{PR}{2} + P_n R \sin\alpha\right) \sin\theta - \left(\frac{P}{2} + P_n\right) \times (1 - \cos\theta) R + P_n R \left[1 - \cos(\theta - \alpha)\right]$$

For the section between E and B_1,

$$\left|\frac{\partial U}{\partial P_n}\right|_{EB_1} = \frac{1}{EI_x} \int_{\alpha}^{\pi/2} M_x \frac{\partial M_x}{\partial P_n} R d\theta + \frac{1}{GK} \int_{\alpha}^{\pi/2} M_z \frac{\partial M_z}{\partial P_n} R d\theta \quad (6)$$

The total deflection is the sum of the two expressions just found, or:

$$\Delta = \left|\frac{\partial U}{\partial P_n}\right|_{A_1 B_1} = \left|\frac{\partial U}{\partial P_n}\right|_{A_1 E} + \left|\frac{\partial U}{\partial P_n}\right|_{EB_1} \quad (7)$$

Performing the necessary substitutions, differentiations and integrations, and equating P_n to zero, the general equation for the deflection of any point on the ring becomes:

$$\Delta = \frac{PR^3}{8}\left[\frac{a}{EI_x} + \frac{b}{GK}\right] \quad (8)$$

where

$$a = (1 + sin\alpha)\ (2\alpha + cos\ 2\alpha - 1) - sin\ 2\alpha\ (1 - sin\alpha) + (1 - cos\alpha)\ (sin\ 2\alpha - cos\ 2\alpha - 2\alpha + \pi - 1)$$

$$b = 2\alpha\ (3 + sin\alpha) - 7\ sin\alpha - sin\ 2\alpha\ (sin\alpha - 3) - cos\ 2\alpha\ (1 + sin\alpha) + 4\ cos\alpha - 2\ sin\ 2\alpha - 3 + (1 - cos\alpha)\ (cos\ 2\alpha - sin\ 2\alpha + \pi - 3)$$

At the particular point when $\alpha = \pi/2$, $a = 2\pi - 4$ and $b = 4\pi - 12$, and the deflection becomes

$$\Delta = \frac{PR^3}{2}\left\{\frac{1}{EI_x}\left(\frac{\pi}{2} - 1\right) + \frac{1}{GK}(\pi - 3)\right\}$$

Also, at the point where $\alpha = O$, $a = O$, $b = O$, and therefore $\Delta = O$.

For practical solution, curves of a vs. α and b vs. α have been plotted, *Fig.* 3. With the aid of these a suitable cross-section can be chosen to yield a predetermined deflection at any point.

MAXIMUM DEFLECTION WITH OFFSET LOAD AND REACTIONS: Sometimes it is necessary to determine the deflection of the central load when it and the reactions at the supports are offset from the center of twist of the ring cross-section. If the central load is offset an amount ϵ from the center of twist, the equivalent loading at B_1 and B_2 is the concentrated load P together with a couple $P\epsilon$ about the Z-axis.

Similarly, if the reactions are offset an amount ϵ_1 from the center of twist, the equivalent loading at A_1 or A_2 is the reaction P together with a couple $P\epsilon_1$ about the Z-axis.

Cutting the ring at A_1 and considering the effect due to the action of P alone, the moments M_x and M_z at any section D at an angular distance θ from OA_1 are:

$$M_x = \frac{P}{2}R\ cos\ \theta - \frac{P}{2}R\ sin\ \theta$$

$$M_z = \frac{P}{2}R\ sin\ \theta - \frac{P}{2}R\ (1 - cos\ \theta)$$

Considering the effect of the couple due to the offset of P, it is evident from the conditions of equilibrium that this couple is resisted at A_1 and A_2 by a moment $M = \pm P\epsilon$ about the X-axis.

In accordance with the moment sign convention, ϵ and ϵ_1 are positive when the offsets are outside the ring; the moments M_x and M_z at section D due to the action of the couple alone are:

$$M_x = \frac{M}{2}cos\ \theta - M_{z_o}\ sin\ \theta = \frac{P}{2}(\epsilon\ cos\ \theta - \epsilon_1\ sin\ \theta)$$

$$M_z = \frac{M}{2}sin\ \theta + M_{z_o}\ cos\ \theta = \frac{P}{2}(\epsilon\ sin\ \theta + \epsilon_1\ cos\ \theta)$$

Applying the principle of superposition, the effective moments at section D due to the combined action of the couple and the load are:

$$M_x = \frac{P}{2}R\left(cos\ \theta - sin\ \theta + \frac{\epsilon}{R}cos\ \theta - \frac{\epsilon_1}{R}sin\ \theta\right)$$

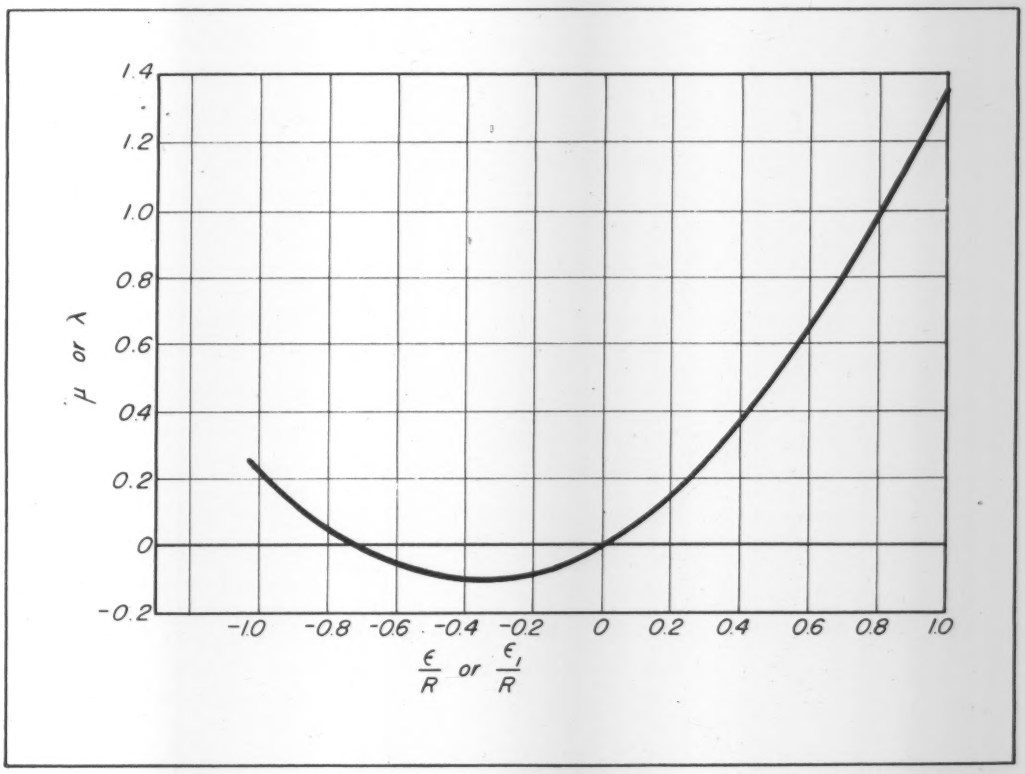


Fig. 4—Curve gives values of the constants to be used in calculating m, Equation 9, when the load is offset from the ring axis

$$M_x = \frac{P}{2} R \left(\sin\theta + \cos\theta - 1 + \frac{\epsilon}{R} \sin\theta + \frac{\epsilon_1}{R} \cos\theta \right)$$

The deflection of B_1 and B_2 along the line of action of P is:

$$\Delta = \frac{\partial U}{\partial P} = 2 \left\{ \frac{1}{EI_x} \int_0^{\pi/2} M_x \frac{\partial M_x}{\partial P} R d\theta + \frac{1}{GK} \int_0^{\pi/2} M_t \frac{\partial M_t}{\partial P} R d\theta \right\}$$

After substitution and the necessary differentiation and integration, the expression for deflection becomes:

$$\Delta = \frac{PR^3}{2} \left\{ \frac{1}{EI_x} \left[\left(\frac{\pi}{2} - 1 \right) + m \right] + \frac{1}{GK} \times \left[(\pi - 3) + m \right] \right\} \quad (9)$$

where

$$m = \frac{\epsilon + \epsilon_1}{R} \left(\frac{\pi}{2} - 1 \right) + \frac{\pi}{4} \frac{\epsilon^2 + \epsilon_1^2}{R^2} + \frac{\epsilon\epsilon_1}{R^2}$$

Comparing Equations 3 and 9, it will be noted that the change in the deflection at B_1 or B_2 is:

$$m \frac{PR^3}{2} \left(\frac{1}{EI_x} + \frac{1}{GK} \right)$$

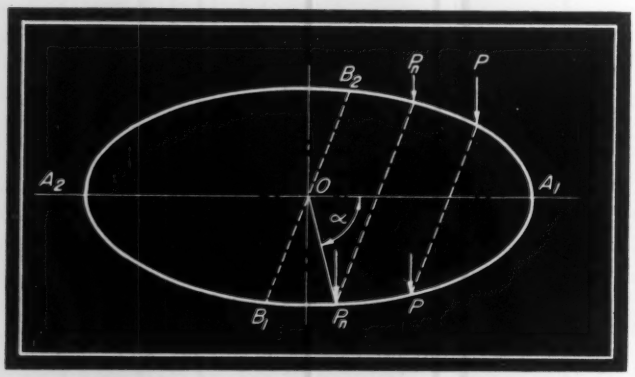
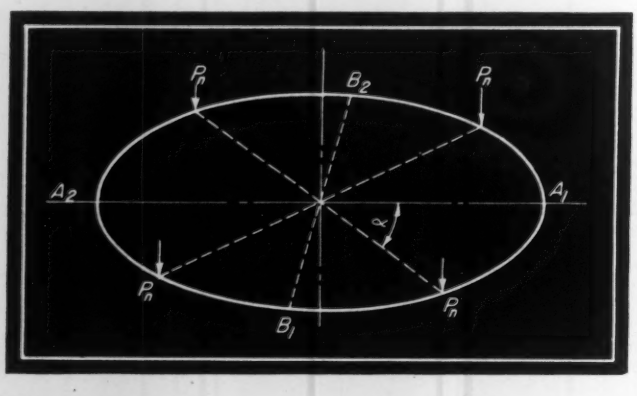


Fig. 5—Above—Offset concentrated loads P cause redundancy, a reactive moment being required for equilibrium

Fig. 6—Below—Circular ring hinged at A_1 and A_2 carries uniformly distributed load around circumference



Calculations may be simplified by rewriting m in the form

$$m = \mu + \lambda + \frac{\epsilon\epsilon_1}{R^2}$$

where

$$\mu = \frac{\epsilon}{R} \left(\frac{\pi}{2} - 1 \right) + \frac{\pi}{4} \frac{\epsilon^2}{R^2}$$

$$\lambda = \frac{\epsilon_1}{R} \left(\frac{\pi}{2} - 1 \right) + \frac{\pi}{4} \frac{\epsilon_1^2}{R^2}$$

Values of μ and λ are plotted in *Fig.* 4. By way of example, if $\epsilon/R = 0.5$ and $\epsilon_1/R = -0.4$ the curve shows $\mu = 0.48$ and $\lambda = -0.12$. Also, $\epsilon\epsilon_1/R^2 = -0.2$, hence $m = 0.48 - 0.12 - 0.2 = 0.16$. The increase in the deflection at B_1 or B_2 is:

$$0.08\, PR^3 \left(\frac{1}{EI_x} + \frac{1}{GK} \right)$$

and the actual deflection at B_1 or B_2 for this condition is therefore:

$$\Delta = \frac{PR^3}{2} \left\{ \frac{1}{EI_x} \left[\left(\frac{\pi}{2} - 1 \right) + 0.16 \right] + \frac{1}{GK} \left[(\pi - 3) + 0.16 \right] \right\}$$

Examining the curve, it is of interest to note that an offset at B_1 or B_2 when the offset at A_1 and A_2 is zero has the same effect on the maximum deflection at B_1 and B_2 as when the offset at B_1 and B_2 is zero and the offset at A_1 and A_2 is ϵ_1 provided $\epsilon = \epsilon_1$. This is also evident from the symmetry of loading. Also, when $\epsilon/R = \epsilon_1/R = -0.72$, $\mu = \lambda = 0$ and $m = 0.52$. Then the maximum deflection at B_1 and B_2 will be increased by

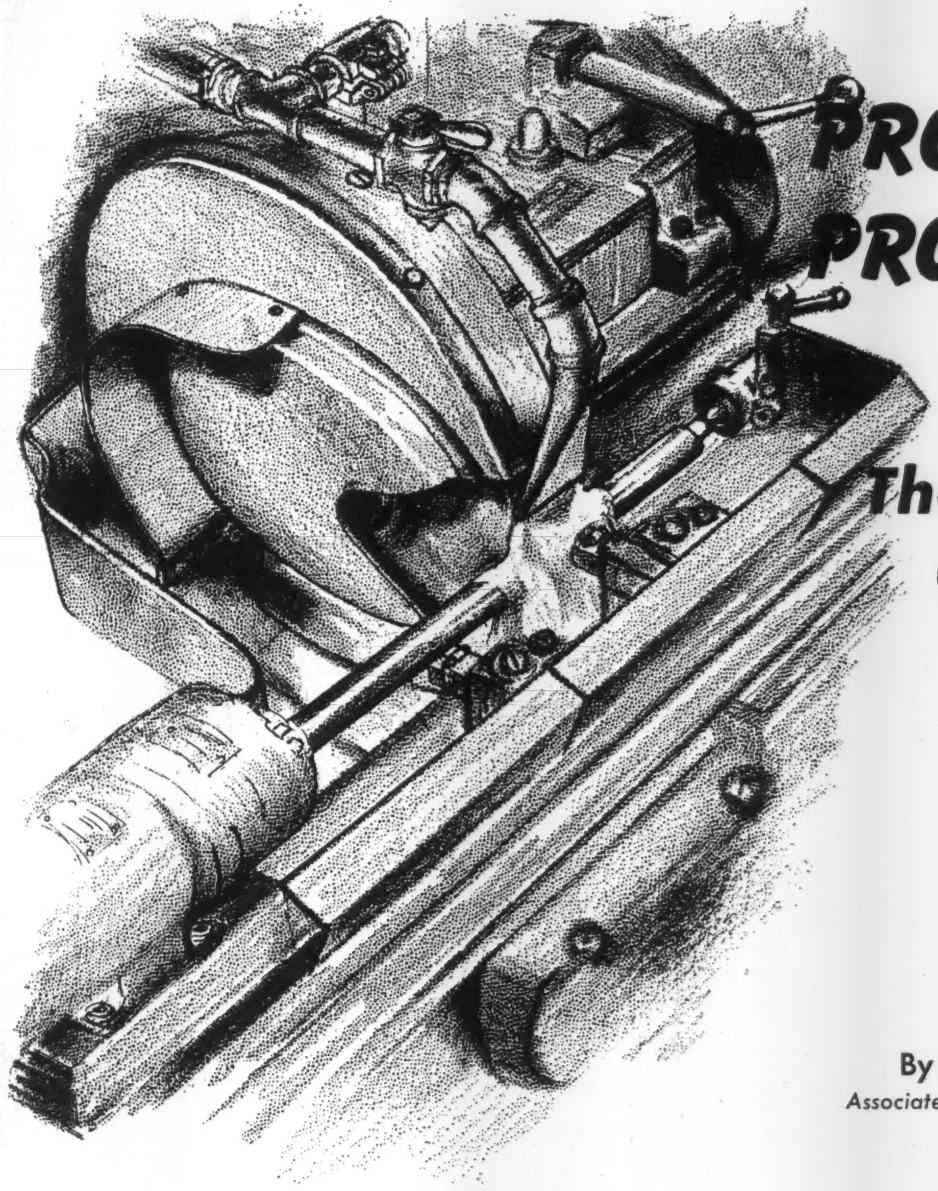
$$0.26\, PR^3 \left(\frac{1}{EI_x} + \frac{1}{GK} \right)$$

If $\epsilon/R = \epsilon_1/R = -1.0$, $\mu = \lambda = 0.215$, $\epsilon\epsilon_1/R^2 = 1.0$ and the maximum deflection at B_1 and B_2 will be increased by

$$0.715\, PR^3 \left(\frac{1}{EI_x} + \frac{1}{GK} \right)$$

UNSYMMETRICAL LOADING: The method of analysis outlined in the foregoing is applicable to any system of externally applied loads so long as the loading is not unsymmetrical. When this condition is not satisfied the element will possess one-degree redundancy, and analysis regarding the deflection of the point of application of any one of the external loads cannot be performed without first determining the magnitude of the redundant reactive element. This can be effected

(*Continued on Page* 196)



PRODUCTION PROCESSES...

Their Influence On Design

By Roger W. Bolz
Associate Editor, Machine Design

Part XXXII–Production Grinding, General Considerations

NEED for improved surface finish as well as facility in finishing parts too hard for ordinary cutting methods resulted in the development of synthetic abrasives around the year 1890. Natural abrasives, which served satisfactorily for centuries in simple sharpening and abrading operations, lacked uniformity and other characteristics required in mass-production grinding and as a result were soon supplanted. Early precision grinding thus had its beginning somewhere around the turn of the century and has since developed rapidly into one of the primary mass-production methods.

Although grinding had been thought of and used extensively as a means for finishing hard surfaces only, abrasive development and grinding machines have progressed until even the softer materials can now be ground efficiently. The much improved finish resulting from the grinding operation, offering surfaces with less tendency toward wear and dimensional change, fast found wide application in machine manufacture and commands the designer's attention in obtaining maximum quality and economy in production of machine parts of all varieties from tiny precision injector nozzles and superprecision ball bearings to gigantic mill rolls.

Not unlike other machining methods, grinding is primarily a process of removing material in the form of chips. Abrasive grains of grinding wheels perform in much the same manner as single-point cutting tools, effecting by a multitude of contacts a virtual multiple-tool cutting action which produces extremely minute chips similar in character to those produced in turning. Since cutting edges of the grits are extremely thin, it is possible to remove much smaller chips and finish surfaces to a much greater accuracy of finish and dimension than with other

methods of metal cutting presently available

Until recently all wheels were dressed or prepared for grinding by using the well-known diamond dressers. Production machines are now largely equipped with automatic dressing or truing devices for wheel preparation. A variety of mechanical form-generating diamond dressers are also available for producing other than plain faces. These, however, are somewhat limited mechanically in the forms which are practicable to produce, and until the perfection of crush dressing technique certain thread forms having angles with the vertical less than 25 degrees could not be produced on multiribbed wheels. In fact it is usually necessary to employ "skip-rib" diamond dressing on multiform wheels, especially on the finer pitches, although this increases the grinding time. With the introduction of the crush dressing technique in production grinding, design possibilities have broadened considerably. A vast variety of complicated forms and shapes can, as a result, be ground or generated from the solid rapidly and accurately. Although machines have been specially adapted for crush-forming operation, many standard machines can be fitted for this method of wheel dressing. Crush dressing, in lieu of traversing a diamond across a grinding wheel to prepare or dress it for grinding, embodies the principle of forcing the abrasive wheel into a crushing roll having a profile complementary to that desired on the wheel.

Simplifying the procedure of dressing an abrasive wheel repeatedly to secure accurate duplication of complex forms, crush dressing thus enhances both quality and quantity in production. Although the somewhat faster, cooler cutting afforded by this method of wheel dressing is an advantage in production of complex forms, certain limitations exist which must be recognized. Extremely deep forms can be crushed but contribute to accelerated crusher wear on long runs, and, too, it is difficult to hold a square shoulder on the wheel especially where the shoulder must be normal to the wheel axis. An advantage over diamond dressing, however, is that worm, Acme, or buttress threads, or similar forms having flank angles within 5 degrees of the vertical, can be readily crushed.

Proper grinding wheel speeds have been fairly well

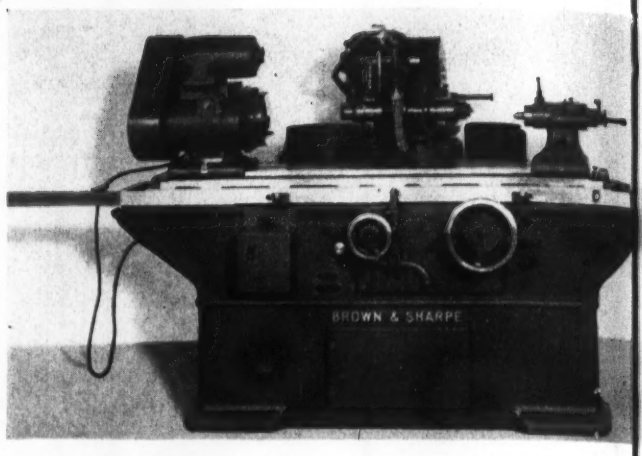


Fig. 2—Standard universal grinder, unlike the plain grinder, allows both headstock and wheelhead to swivel

Fig. 3—Below—Thread and form grinder which offers either diamond or crush dressing as required

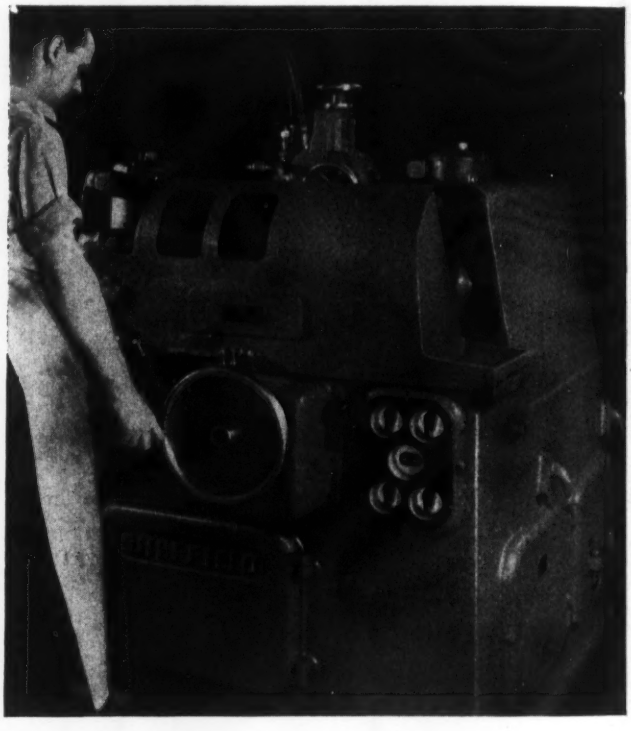
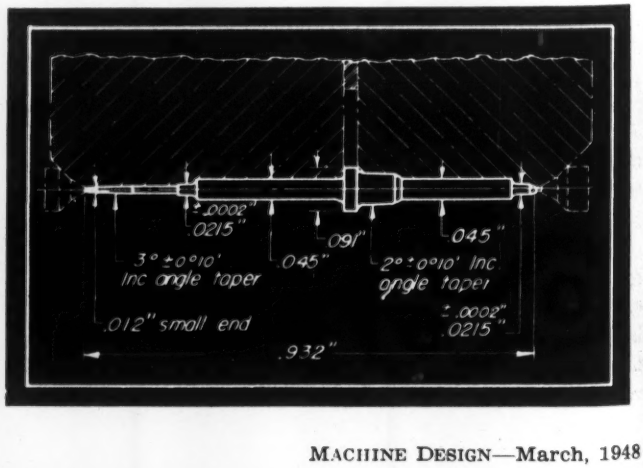


Fig. 4—Instrument shaft center-ground on a small form grinder. Diamond dressing is used for delicate details

Fig. 1—Standard plain grinding machine. Axis of workpiece can be angled for grinding tapers



established and most machines available can be readily adjusted to attain the desired cutting efficiency. Major problem in the past has been with internal grinding where proper peripheral wheel speeds were difficult to obtain. Today, however, high-cycle, direct-driven wheel heads are rapidly supplanting previous types. Efficient peripheral wheel speeds observing maximum manufacturers limits for specific wheels, of course, are as follows:

Cylindrical grinding	(resinoid)	5500 to 12,000 sfpm
	(vitrified)	5500 to 6500 sfpm
Surface grinding		4000 to 5000 sfpm
Internal grinding		2000 to 6000 sfpm
	(diamond wheels)	5000 to 6500 sfpm

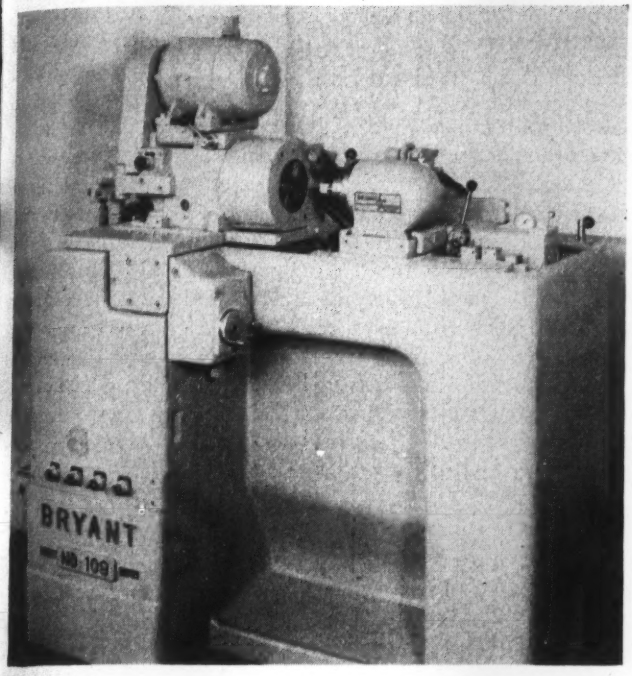
For obtaining the desired finish as well as maximum stock removal commensurate with good wheel life, proper work speeds are also necessary. Except for internal centerless grinding, at the line of cutting the work and the abrasive wheel pass in opposite directions. Work speed at this point should vary somewhat according to the material being ground, approximately as follows, the low range being used for roughing cuts and the high for finishing:

Soft steel	30 to 50 sfpm
Hard steel	70 to 100 sfpm
Cast iron	200 to 400 sfpm
Aluminum	100 to 200 sfpm

Most efficient stock removal is obtained with coarse-grit wheels and while surface finish is distinctly rougher, commercially acceptable dimensional accuracy can be maintained. Maximum economy in manufacture thus depends to a large extent upon judicious selection of surface finish. The following figures on stock removal per single pass indicate generally the comparative rates, though these will vary somewhat for different materials:

Rough grinding	0.002 to 0.020-inch
Semifinish grinding	0.001 to 0.010-inch
Finish grinding	0.0005 to 0.005-inch

Particular design features of a part dictate to a large extent the type of grinding machine required. Material specified and its condition indicate, as a



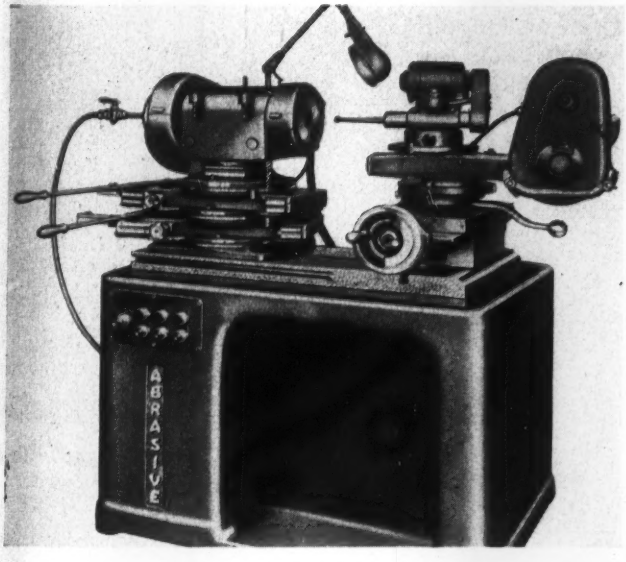


Fig. 5 — Top — Standard automatic chucking grinder for parts up to 9 inches in diameter. Grinding stroke is 4 inches and wheel speeds up to 100,000 rpm are possible

Fig. 6—Above—Internal finisher with compound workhead slides to enable grinding of two angularly related holes at one setting

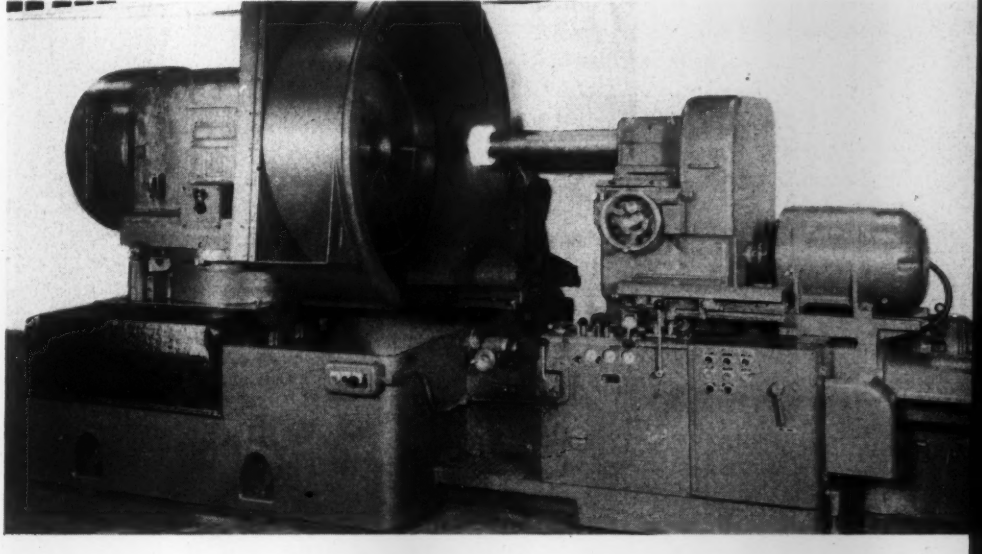


Fig. 7—Right—One of the largest of internal grinders, this model will swing 60 inches and grind to a depth of 16 inches

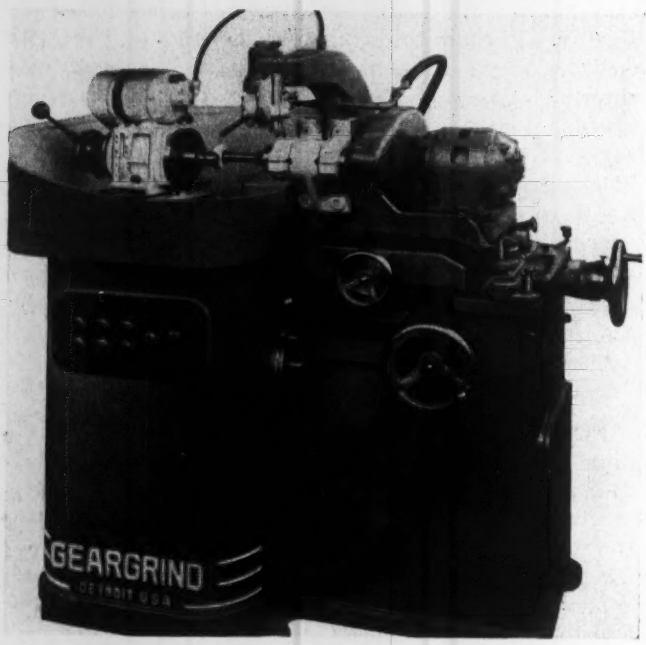


Fig. 8—Universal oscillating grinder widely used for finishing bearing races, spherical and similar parts

rule, the most desirable type abrasive wheel. Where processing costs are excessive, redesign to utilize a less expensive, higher output grinding method may be found well worth while. For instance, wherever possible the production economy of surface, disk and centerless grinding should be taken advantage of by proper design consideration.

TYPES OF MACHINES AND THEIR CHARACTERISTICS: Today, with but few exceptions, practically any reasonable grinding operation can be handled rapidly and economically, providing due consideration is given the design of the parts to be ground. Where projections are eliminated and surface area brought to a minimum, the surface grinder can be used to advantage in removing the smallest possible thickness of material to attain a true surface. Again, by proper design the speed and accuracy of the centerless grinder can help reduce machine parts costs significantly. Of greatest importance, where quantities to be produced do not justify purchase of a new machine, adequate knowledge of the machines at hand will make possible the design of functionally and economically satisfactory parts.

Cylindrical Grinders: The identifying term "cylindrical" is applied to that large group of grinding machines which employ centers for mounting the workpiece to be ground. As with the automatic lathe two centers are used, a dog or other attachment driving the workpiece. Where desirable, collets, chucks, or other holding devices can be substituted. For the most part grinding is confined to simple external cylindrical surfaces, tapers, faces, etc.

Most widely used of the external cylindrical machines is the so-called **plain** grinder which is a general-purpose production machine for the grinding of axles, shafts, spindles, etc., *Fig.* 1. Plain grinders are built with hydraulic, mechanical, hand table traverse, or a combination of all three, and feature various types of wheel and work drives such as electronic, electric, etc. Work table, with power headstock and tailstock which can be positioned to suit, is mounted on the longitudinally traveling main table so as to allow angular adjustment relative to the table ways up to around 10 degrees. Angular adjustment of the swivel table varies with different size machines. Cross feeds or wheel infeeds can be continuous or interrupted and are hand powered, independent automatic (with the table stationary) or full automatic (with the table traversing and a predetermined infeed at table reversal) or a combination of all three. The entire grinding cycle can be manual, semiautomatic or full automatic as desired. Plain grinders can handle parts from the smallest up to 36 inches in diameter by 192 inches in length, the lengths being proportioned to the swing capacity of each machine. A 3-

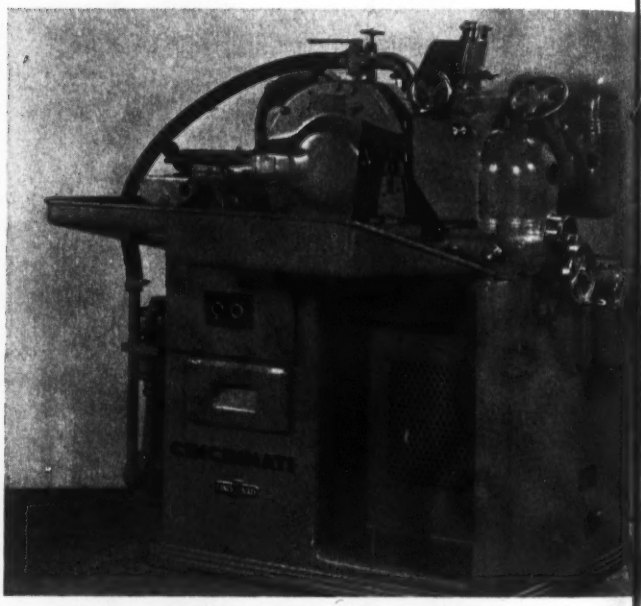
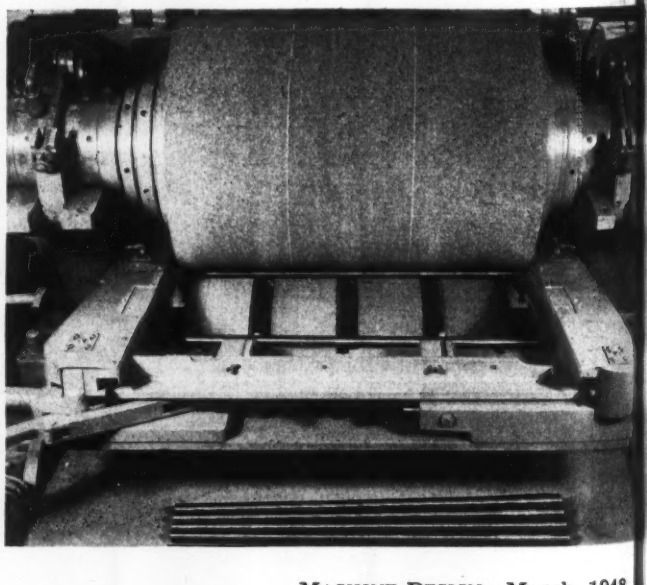


Fig. 9—Centerless grinders offer a range of production possibilities and outstanding economy

Fig. 10—Below—Rifle barrels produced by in-feed or plunge grinding by centerless method



inch or 4-inch machine is available in 12-inch or 18-inch lengths, a 6-inch machine in 18 and 30-inch lengths, a 10-inch machine in 18, 36, 48, 72, 96 and 120-inch lengths, a 14 or 16-inch machine in 36, 48 and on up to 168-inch lengths in 2-foot increments, while 20-inch and larger machines generally start at 96 inches and go up to 192 inches in 2-foot increments.

Second most widely used machine is the so-called **universal** grinder which, though originally a tool-room machine, is now also a useful production unit, *Fig.* 2. In addition to the features offered by plain grinders, universal machines are provided with a swiveling headstock and a swiveling wheelhead. Completely flexible in setup, these machines are ideally suited for production of small or large quantity lots of parts requiring face as well as cylindrical and taper grinding. Complex parts are readily handled and where necessary internal grinding can also be performed. Universal machines are available to handle parts requiring swing up to 18 inches and center distances to 72 inches.

Another group of machines which fall under the external-cylindrical category are the **thread** and **form** grinders, *Fig.* 3. Certain of these grinders are equipped for both single and multiform wheel operation and have both diamond and crush dressing attachments. As a rule crushing can be used successfully for thread forms up to about 40 pitch which requires a 400-grit abrasive in order to maintain the necessary crest and root dimensions. However, 72-pitch threads have been ground experimentally with a 600-grit wheel, crush dressed. Diamond dressed wheels can be single or multiribbed up to about 80-pitch and as a rule diamond dressing is recommended for special designs and forms requiring sharp angles, corners, extra fine detail or minimum possible tolerances, *Fig.* 4. Machines are provided with both manual and/or automatic operational cycles and are designed to handle a wide variety of thread and form work up to about 10 inches in diameter and about 6 feet long.

Although most of the foregoing external-cylindrical machines can be fitted with internal grinding attachments, the grinders are designed primarily for external work. Another group of grinders, specifically designated as **internal** or **chucking** grinders, *Fig.* 5, are available for economical production finishing of holes in gears, bushings, bearings, etc., or more complex bores in precision machine parts, *Fig.* 6. Generally these grinders are equipped with a reciprocating wheel head and a rotary work head which is mounted on a movable cross slide with provision for angular adjustment. Face plates, chucks, collets or fixtures can be used for holding the work. Adaptable for both manual or full automatic operation, certain machines feature in addition automatic rough and finish grinding, automatic dressing and sizing. Standard machines provide wheel speeds up to 100,000 rpm and will finish holes from 0.040-inch to 50 inches in diameter. Straight, tapered, curved and irregular holes or a combination of these can be handled as well as face grinding where squareness is desirable. While standard machines range up to 60 inches swing allowing a maximum hole of 50 inches diameter by 16 inches deep, *Fig.* 7, specials are in service providing swings up to 100 inches.

Completing the general category of cylindrical grinders is the group known as universal **oscillating** grinders. Provided with a wheel head having reciprocating as well as cross-feed motion, these grinders can perform a variety of both external and internal cylindrical or conical finishing operations, *Fig.* 8. For such operations the oscillating workhead can be locked in position. For finishing internal or external spherical surfaces of parts, the wheelhead can be held stationary and the workhead can be oscillated through 180 degrees by hand, or power oscillated from zero to 90 degrees. Work up to 9 inches O. D. and $\frac{3}{8}$-inch to 8 inches I.D. by 6 inches in length can be handled in collet, chuck, faceplate, or fixture.

Centerless Grinders: As contrasted with regular cylindrical grinders, centerless machines develop cylindrical or formed parts without the centering operation and as a consequence have far greater output, *Fig.* 9. Actually, only one-half as much stock need be removed to produce a complete finished surface as in center grinding. Usually centerless machines utilize a large grinding wheel operating at 6000 to

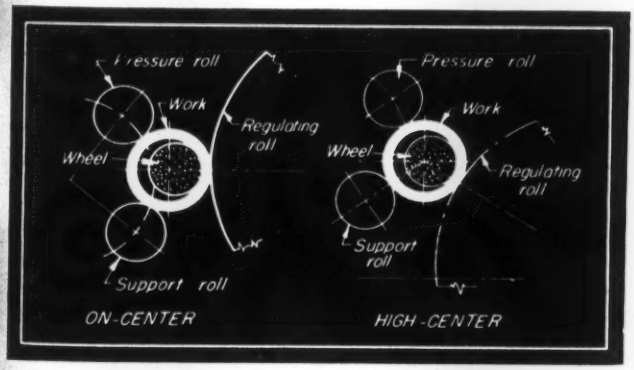


Fig. 11—Above—Principles of operation in centerless methods for internal grinding

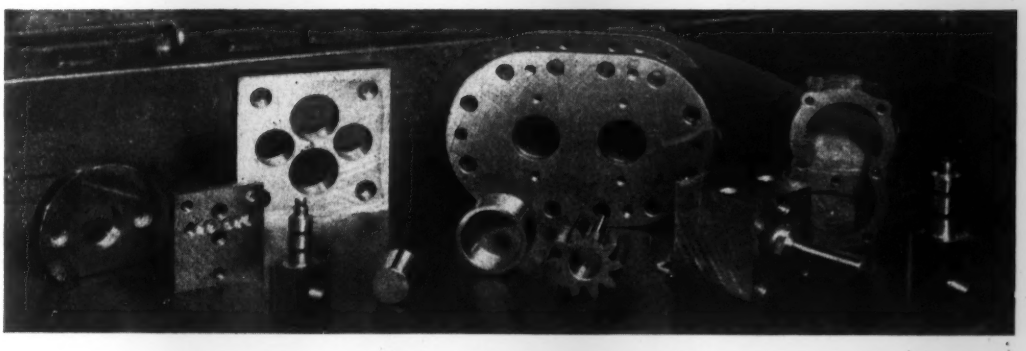


Fig. 12—Right—Variety of precision pump parts all surfaced on a vertical spindle machine

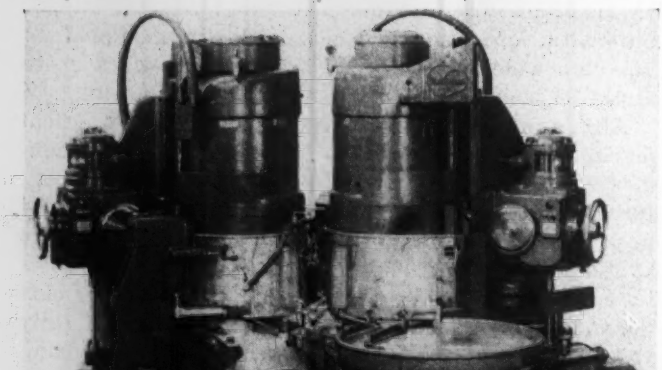


Fig. 13—Multiple-spindle automatic surface grinder for high-production grinding

8000 sfpm, a rubber-bonded abrasive regulating wheel rotating in a like direction at about 40 to 1200 sfpm, and an adjustable work rest or support. Contact on the slow-moving regulating wheel of the piece being ground maintains uniform rotational speed of the work in a direction opposite that of the grinding wheel. Several classes of centerless grinding are normally employed depending upon the type of work to be processed. Straight, cylindrical surfaces having no interfering shoulders or projections are ground by the **through-feed** method and machines available can handle diameters from about 0.020-inch up to about 7 inches and 20 feet or more in length. **Abrasive belt centerless** grinders are available which handle work up to about 9 inches in diameter. Straight threaded parts or shafts from 8 pitch to 80 pitch, ⅛-inch to 36 inches or longer and up to 5 inches O.D. can be rapidly ground on **centerless thread** grinders. Whether parts are long or short, through-feed grinding is continuous and production extremely high.

For formed or contoured pieces, parts having shoulders, heads, flanges, or designs requiring undercuts or tapers, **plunge** or **in-feed** centerless grinding, in which the work does not traverse relative to the wheels is employed, *Fig.* 10. Many automatic controlled-cycle machines are available with diamond and/or crush dressing attachments for a wide variety of part designs. Maximum obtainable plunge movement of the grinding wheel, however, is about 1-inch and parts cannot be produced having radial variations greater than this amount. Part length is limited by wheel width, available equipment will usually accommodate forms up to 36 inches wide and diameters up to 7 inches. Depending upon the size and design, production can range up to 1000 pieces per hour.

Used only for special taper work, in **end-feed** centerless grinding the part is fed manually or mechanically between the wheels to a fixed stop, and ejected. Threaded parts with shoulders or heads can often be handled by this method.

Somewhat similar to external centerless grinding is that known as **internal centerless** grinding. Having the production advantages of ordinary centerless machines, internal machines allow the rough and finish grinding of inside surfaces of straight, tapered, interrupted, or blind holes with minimum chucking and handling. **On-center** method gives maximum support to the work and makes possible accurate grinding of thin wall sections without distortion while **high-center** method provides for accurate finishing of parts with slightly varying outside diameters or multiple work, *Fig.* 11.

Surface grinders: There are two basic types of surface grinders—horizontal and vertical. In the **horizontal** surface grinder the wheel axis is horizontal and grinding is done on either the face or the periphery of the wheel, while **vertical** grinders have a vertical wheel axis with the grinding taking place on the face of a cylinder or segmented-cylinder wheel. Each type is available with rotary or reciprocating-table.

Rotary table machines are adapted primarily to rapid production of flat surfaces on small and medium-size parts singly or continuously, *Fig.* 12. Vertical machines are capable of carrying segmented wheels up to 42-inch diameter and, with a 60-inch magnetic rotary table chuck, will handle work up to 96 inches in diameter. Vertical-spindle rotary machines are available also in multiple-spindle models for rough, semifinish and finish grinding at one pass with automatic sizing and wheel advance as well as part gaging, *Fig.* 13. Reciprocating grinders are pri-

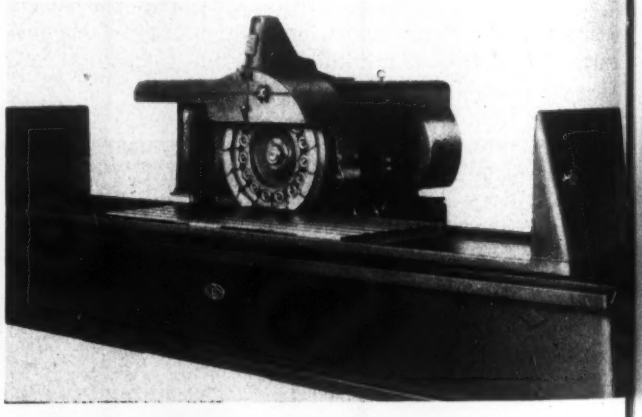
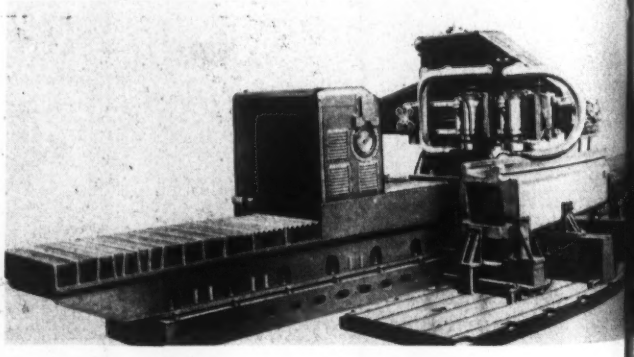


Fig. 14—Above—Reciprocating table horizontal spindle machine for surface grinding large parts

Fig. 15—Below—Precision grinder which develops flat surfaces for any combination of machine slide way designs



marily designed for producing long, straight surfaces of either flat, *Fig.* 14, or formed section, *Fig.* 15.

Also important among the production surface grinders are the **disk** grinders. These are available in horizontal-spindle models with single or opposed spindles. Single vertical-spindle machines available are used mainly for surfacing flat areas without maintaining particular dimensions. Utilizing large, solid abrasive disks or cylinder wheels, horizontal single or opposed-spindle machines have oscillating, reciprocating, rotary or through feeds and are ideally adapted for the production of parts requiring parallel, flat faces such as piston rings, springs and bearing races, *Fig.* 16. Automatic work sizers are provided on many models and operation can often be fully automatic, *Fig.* 17. Work up to 14 inches in length can be ground in opposed-wheel grinders with wheels up to 30 inches.

A special form of flat surface grinder, the **abrasive belt** machine, has been covered previously in Part XVI of this series. Naturally, many special production adaptations of the various types of grinders discussed in the foregoing have been made but such applications require long runs in mass production quantities to be practicable. Again there is a great variety of such machines as **roll** grinders which are capable of grinding all types of rolls up to 60 inches in diameter and 24 feet in length with straight bodies or from ¼-inch concave to ¼-inch crown.

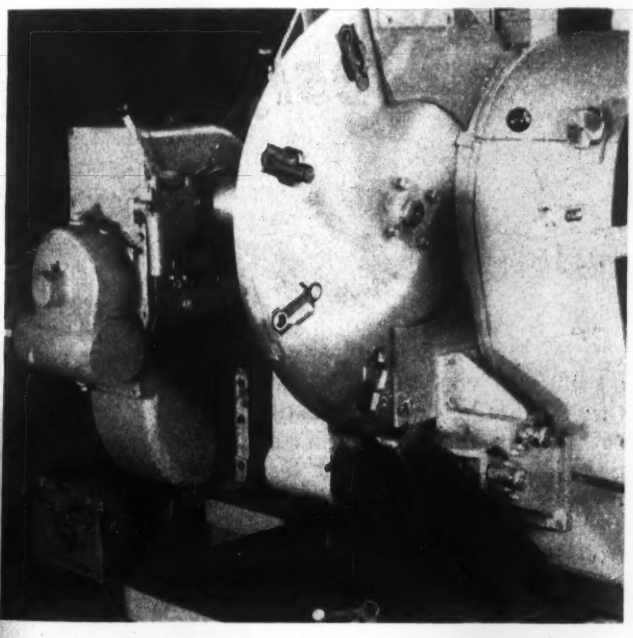
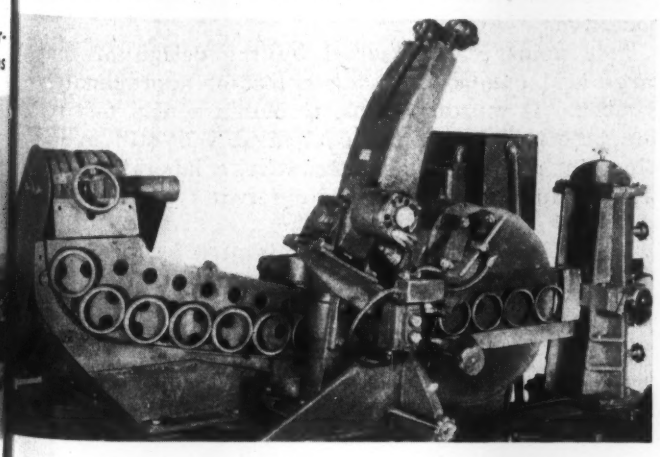


Fig. 16—Above—Opposed-spindle disk grinder with a rotary fixture for grinding small connecting rods

Fig. 17—Below—Fully automatic disk grinder for ball bearing races with one wheel retracted to show path of travel



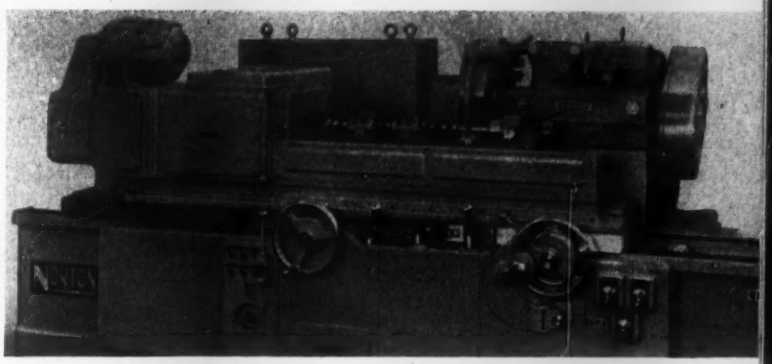


Fig. 18—Typical automatic cam grinding machine for producing single or multiple cams

Also available are **crank-pin** grinders for finishing compressor and engine crankshafts. Machines carrying up to 9 wheels can grind, for instance, seven line bearings, fan pulley and gear fits, and O.D. of the flange in one operation. At the other extreme, using three wheels, center bearing, end line bearing and flange O.D. can be ground.

Single and multiple automatic **cam** grinders, *Fig.* 18, are available for finishing to size all varieties and combinations of cam shapes. Atachments for cam grinding can be used to convert many standard cylindrical grinders for this purpose also.

Too, a variety of **gear** grinders can be utilized in producing precision gears of all designs. Although small or medium quantities of certain gears can be produced on horizontal-spindle surface grinders with special indexing attachments, large-quantity production indicates a standard gear grinder for maximum economy and accuracy. Much the same is true of **splineshaft** grinding.

Next article in this series, Part XXXIII—Production Grinding, Design Factors, will deal with specific considerations in design for grinding, characteristics and suitability of materials and manufacturing tolerances.

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Ex-Cell-O Corp. (*Fig.* 4) ..... Detroit, Mich.
Gear Grinding Machine Co. (*Fig.* 8) .... Detroit, Mich.
Giddings & Lewis Machine Tool Co. (*Fig.* 15) ..... Fond DuLac, Wis.
Hanchett Mfg. Co. (*Figs.* 16 & 17) .. Big Rapids, Mich.
Heald Machine Co. (*Fig.* 11) ..... Worcester, Mass.
Landis Tool Co. ..... Waynesboro, Pa.
Mattison Machine Works ..... Rockford, Ill.
Norton Co. (*Fig.* 18) ..... Worcester, Mass.
Sheffield Corp. (*Fig.* 3) ..... Dayton, O.
Thompson Grinder Co. ..... Springfield, O.

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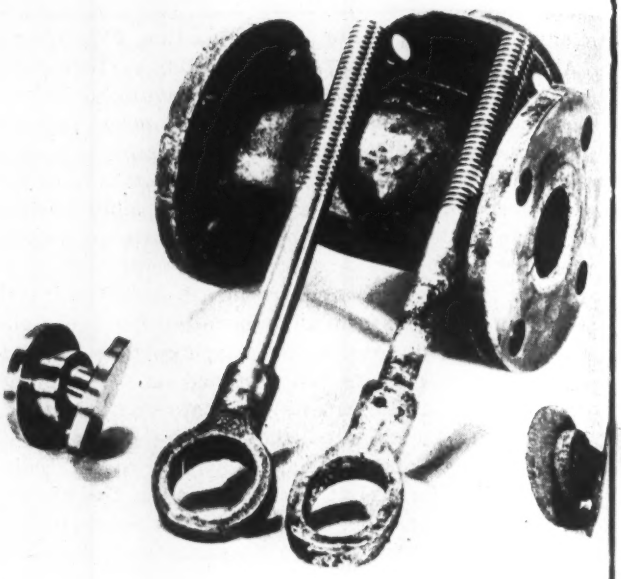


How To Combat Corrosion through Design

Part II—Design Measures

RESULTS of corrosion tests when properly interpreted can be effective tools in design, serving as useful guides for correct materials selection. In the majority of tests, the specimen area is measured and the specimen is weighed before and after testing to obtain weight loss per unit area. The specific gravity of the material in question is usually taken into consideration to secure results indicating depth of penetration per unit time. Of this type, inches penetration per year is one of the most common methods of data presentation. The main factor of importance to be understood is that the result of a corrosion test involving weight loss is not necessarily an indication of the suitability of a material for a machine part application.

In short, the results cannot be used as data for a method to actually calculate the predicted life of a machine or structure. If the attack were always uniform and overall, this would be possible, but unfortunately, this is rarely the case. Such actions as are incurred in "concentration cells" and "pitting" usually make localized attack the rule instead of the exception.

If it is possible to obtain them, field corrosion tests or pilot plant tests are the best sources of information. Through these, the test specimens are subjected to the actual environment in which the prospective machines will operate. Unfortunately, in many cases these are inconvenient, take too much time or are impossible to conduct. The next best results would then be obtained from laboratory tests. These may be of two types: (1) Tests simulating actual conditions and (2) accelerated tests. The former is the better of the two, however, the validity of the results would, of course, depend upon the differential between the actual and simulated conditions. Results from accelerated tests should be diagnosed warily. For practical application they must be produced by an acceleration brought about only by intensifying the controlling factors of the environment. However, in most cases this is not done as, due to the nature of the acceleration, new variable factors are introduced which are not included in the operating conditions.

This point is emphasized by the design of four large acid pumps, involving a cost of approximately $14,000. It was impossible to obtain a field test for the process because the equipment was still in the blueprint stage. As an alternative, a laboratory test was conducted, consisting of different types of metal

TABLE III
Static Corrosion Test

Metal	Corrosion Rate (I.P.Y.)*
Bronze A	0.03
Stainless A	0.30

* Inches penetration per year.

Left—Wrong type stainless steel for valve parts resulted in corrosion shown at right. Correct type of stainless (left) did not corrode

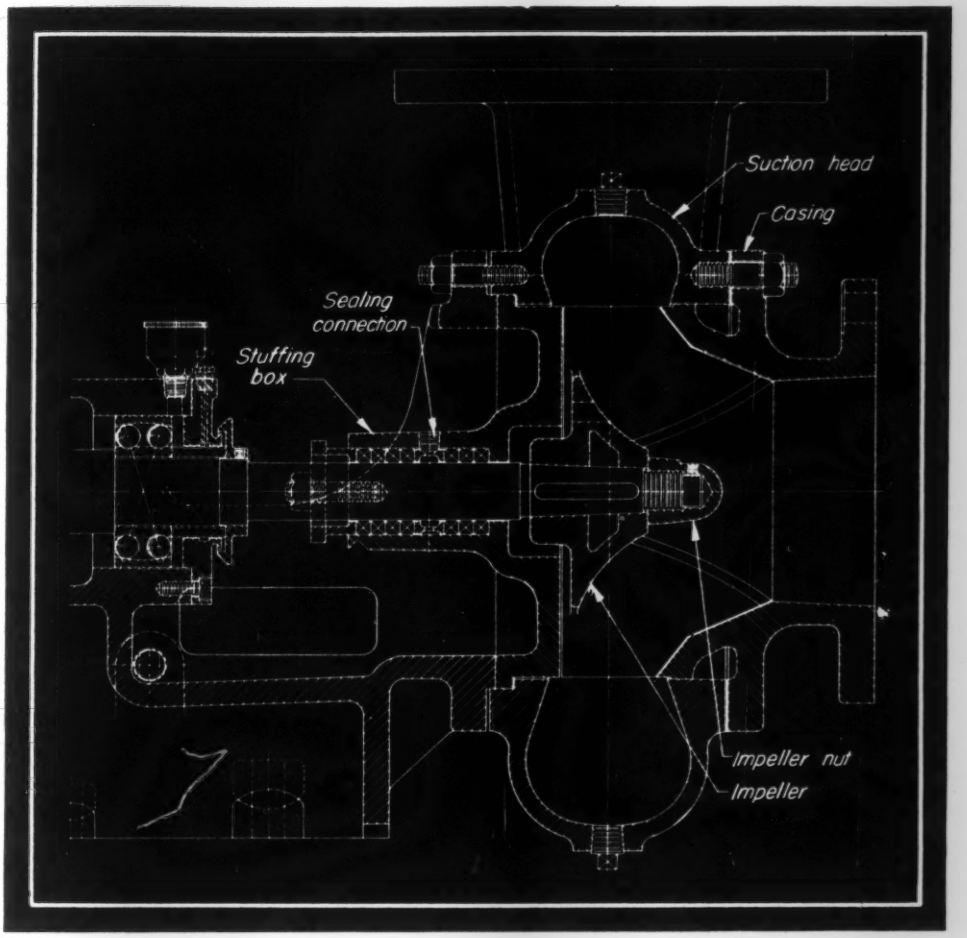


Fig. 11 — Right — To prevent galvanic corrosion in this pump, all parts in contact with the electrolyte are made of the same type of material, i.e., cast iron or steel

being immersed in the solution which was to be pumped. The results, given in TABLE III, indicated that bronze should be the material for design.

These results were suspected by the pump manufacturer, especially after obtaining the case history of the test. Consequently, the same metals were subjected to a laboratory test developed by the manufacturer. Samples were suspended in the solution and were revolved at constant speeds up to 45 feet per second. This final speed approached the velocity that would exist inside the pumps. Results of this test are given in TABLE IV. As can be seen, the static tests were completely misleading. The bronze, which was considered the best by the first test, increased in corrosion rate with velocity to such proportions as to make it unfit for the application. Apparently, the protective film of the bronze was detrimentally affected by the higher velocities to which it would be subjected with a pump application. With all factors taken into consideration, the chrome-nickel-molybde-

TABLE IV

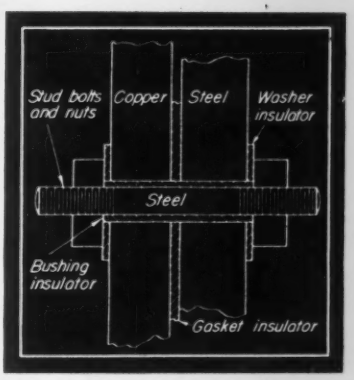
Agitated Tests Corrosion Rates (I. P. Y.)*

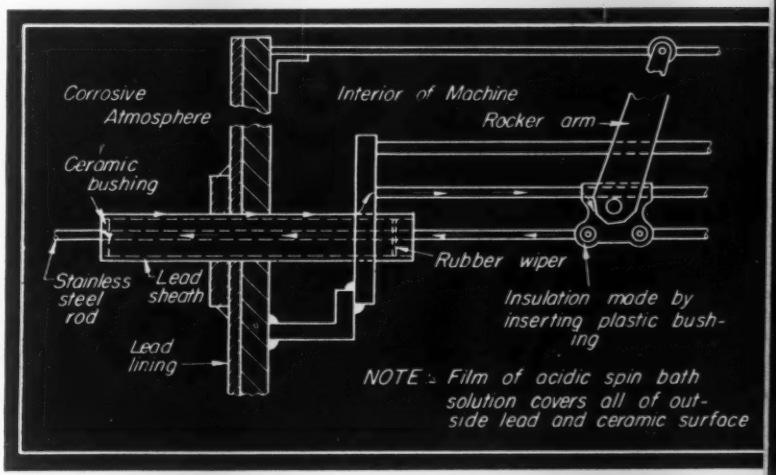
Metal	Solution Velocity 5 ft/sec for 24 hr	5 ft/sec for 72 hr	40 ft/sec for 7 hr
Bronze A	0.028	0.600	0.464
Bronze B	0.027	0.642	0.860
Bronze C	0.036	0.816	0.423
Stainless B	0.081	0.058	0.059
Stainless A	0.103	0.020	0.019

* Inches penetration per year.

Fig. 12—Right—Galvanic corrosion can be prevented by insulating parts

Fig. 13—Below—Arrows indicate path of current responsible for galvanic corrosion in machine operating in a saturated sulfuric-acid vapor environment





num alloy (stainless A) was selected.

Another illustration of this point: Experience proved to the same pump company that their complex chrome-nickel-molybdenum pumps and valves usually gave a longer life in hot sulphuric-acid service than could be predicted by the static immersion test commonly used. Hence laboratory test equipment had been designed to subject alloy specimens to high velocities such as occur in pipe lines, through valves and inside of centrifugal pumps. The comparison of corrosion rates of this alloy by air agitated (practically static) tests and by mechanically agitated tests (high velocity) explained why it gave longer service than could be predicted by the former relatively static laboratory tests. The mechanically agitated tests, at 5 ft/sec. velocity, closely simulate conditions in the usual pipe line. Higher velocities do not give appreciably different results on stainless and high chrome-nickel-moly alloys, but the higher velocities do greatly accelerate corrosion on the various bronzes.

Complete Test Data Required

It readily can be seen from the foregoing that the qualitative conclusions are as important as the quantitative ones. The complete case history of the test should be known and understood by the person making the material selection, with not only the weight loss being considered, but surface appearance as well. If correctly interpreted, corrosion tests can be of valuable assistance to the designer for material selection and in the design itself. In many environments the latter factor assumes the major role.

"Direct chemical corrosion," defined in Part I, is exemplified by the corrosion of iron by atmospheric oxygen at elevated temperatures. Here the main problem is that of material selection, and consideration of each case with respect to the wide selection of alloys formulated for resistance to high temperatures should give a satisfactory solution to most such problems.

When galvanic corrosion is encountered, the "galvanic series", TABLE II, Part I, presents a good indication as to what type of material might be used for the component parts of a machine. The most obvious method of eliminating galvanic corrosion is to use one metal or alloy in the entire construction of a machine. A pump to transfer a highly ionized electrolyte, as illustrated by *Fig.* 11 is a good example of this type of design. The casing, suction head, bolting, impeller, impeller nut, setscrew, shaft, and in fact all of the parts in contact with the electrolyte, are of cast iron or steel.

Unfortunately it is not practical in the majority of cases to construct a machine of only one type of alloy. If a bronze impeller nut were substituted in the pump described, there would be galvanic action, as indicated by the galvanic series, with the bronze nut acting cathodic to the iron. However, the current density would be so small, due to the small cathode-anode ratio that no harm would be done to the adjacent parts. On the other hand, the threads on the steel shaft might be affected if the sealing were not perfect, causing a consequent loosening of the nut. To be safe then, the nut should be of the same material.

If the pump were entirely constructed of bronze with the exception of an iron impeller nut, the cathode-anode area ratio would be very large. This would cause a rapid destruction of the nut due to the high current density that would be produced on the anodic area. This is the type of design to strictly avoid.

Included in the same category would be the rapid corrosion of steel rivets in a copper plate (steel is above copper in galvanic series). Conversely, corrosion of copper rivets in a steel plate would be negligible. If it were absolutely necessary to use steel

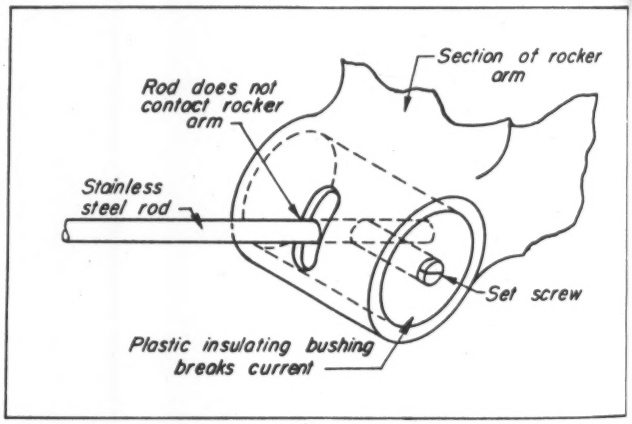

Section of rocker arm
Rod does not contact rocker arm
Stainless steel rod
Set screw
Plastic insulating bushing breaks current

Fig. 14—Above—Use of plastic bushing in rocker arm of machine shown in Fig. 13 disconnects current path and prevents galvanic corrosion

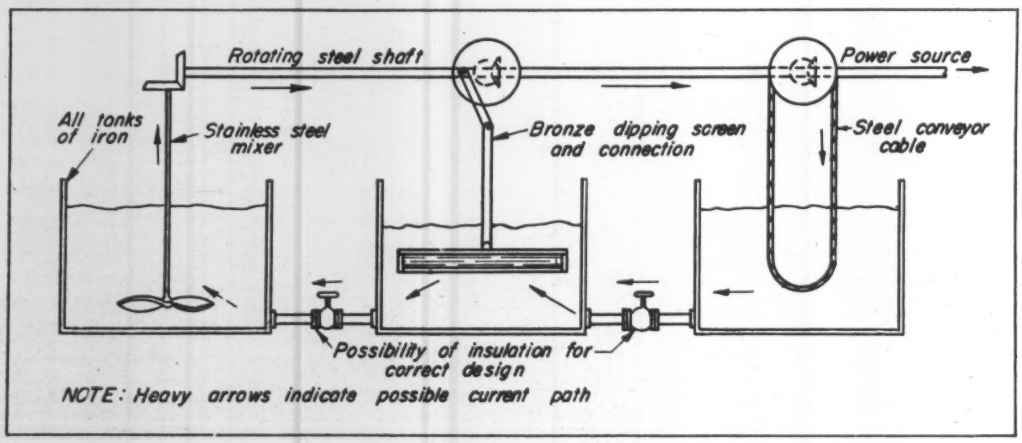

Rotating steel shaft
Power source
All tanks of iron
Stainless steel mixer
Bronze dipping screen and connection
Steel conveyor cable
Possibility of insulation for correct design
NOTE: Heavy arrows indicate possible current path

Fig. 15—Left—How use of different metals and different electrolytes in semicontinuous processing machine can set up "metal-ion cell" type of corrosion. Insulation at points indicated would correct the condition

Fig. 16 — Right — Even though all parts of this processing machine are steel, electrical potential is developed because liquids in tanks are different, and corrosion is incurred

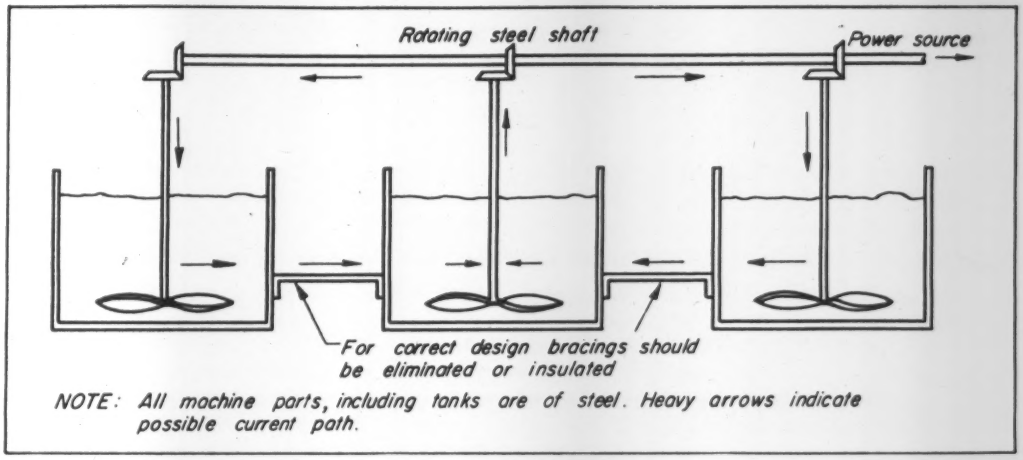
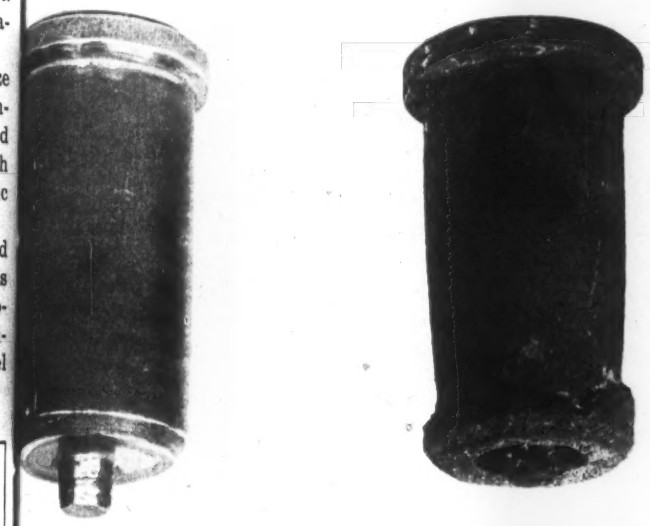

Rotating steel shaft
Power source
For correct design bracings should be eliminated or insulated
NOTE: All machine parts, including tanks are of steel. Heavy arrows indicate possible current path.

Fig. 17—Below — Steel valve stop plug shown before (left) and after (right) subjection to severe oxygen-concentration-cell corrosion



rivets in a copper plate, artificial protection may be employed. This would consist of placing zinc washers under the rivet heads. As indicated by the galvanic series, the iron rivets in the copper plate would then act cathodic to the zinc washers, which would have to be periodically replaced.

Since galvanic corrosion is the result of an electrical current, it is logical to assume that it might be eliminated by disconnecting the circuit. This actually can be done by insulating the two metals in question. A simple example of a joint insulated in such a manner is illustrated by *Fig.* 12. It is to be remembered that while insulation of the members will stop galvanic corrosion, it will in no way affect corrosion due to "direct reaction" of the individual metals by the corrosive medium.

In many cases galvanic corrosion can be reduced or prevented by application of surface coatings to the metal. Paints and lacquers, as well as sprayed metallic, hot-dipped, or electroplated coatings are good examples of this type of protection. It is to be emphasized that covering only the anodic metal is poor design. If this is done and a break or porosity occurs in the coating, a destructive current density would result. This would be due to the large cathode-anode ratio created. It is better to cover the cathodic surface, if only one of the metals is to be considered. However, covering both the anodic and cathodic areas is preferred design.

This type of insulation, with reference to machine design, would not be suitable with an application as shown in *Fig.* 12. However, it is used to advantage in a spray pickling machine, which uses as its pickling agent 15 to 25 per cent sulphuric acid at approximately 160 F. This is recirculated by two nickel-chrome-molybdenum steel centrifugal pumps which are integral parts of the machine. The machine is lead lined and fitted with lead piping. It was found by test that with the described environment, which caused de-aeration of the acid and activation of the alloy material, it was possible for the lead and the alloy steel to migrate in the galvanic series so as to position the stainless steel above the lead. This caused the alloy steel to be anodic to the larger lead surface, introducing the cathode-anode area effect, and producing galvanic corrosion on the pumps. A casing of one of these units so affected, is illustrated by *Fig.* 8, Part I. This particular problem was easily solved by the addition of an oxidizing agent to the acid which "passivated" the alloy steel, causing it to relocate in the galvanic series to a position below that of lead. However, the galvanic action could have been eliminated by insulating the pumps from the rest of the system. This could be done by insulating the pump flanges and bolting from the piping, as indicated by *Fig.* 12.

Surface Film May Cause Galvanic Corrosion

Galvanic action does not require that the dissimilar metals be submerged in an electrolyte. Just a surface film on the metals so connected is sufficient to produce this action. For example, a machine was designed to enable a reciprocating action to be obtained in a saturated sulfuric acid vapor environment. The form of this machine was of structural steel, and the surface, being in direct contact with the vapor, was lead-lined. The reciprocating rods were stainless steel, and were connected to an internal rocker arm. A section sketch of the machine is illustrated in *Fig.* 13.

The stainless steel rods failed due to galvanic corrosion. While, from the front of the machine, the

two metals were not apparently galvanically connected, investigation proved that internally the design was such as to produce a galvanic effect with the sulphated lead being cathodic to the rods. The current path is indicated by the arrows. The correct design necesary to eliminate this action was simple, necessitating only insulation of the two members. This could be done as illustrated in *Fig.* 14. As will be noted, the insulating bearing bushing interrupts the current, eliminating the galvanic action. It is to be emphasized that the above examples are exceptions, not the rule. In all but a few cases lead and stainless steels are satisfactory when connected. However, certain environmental conditions make this combination dangerous and should be recognized. The examples cited could hold true with any two metals.

An example of "metal-ion cell" corrosion (defined in Part I) in which different electrolytes and different metals are involved is the machine sketched in *Fig.* 15. Here three different operations are performed in a semicontinuous process. All action is obtained from one power source, and the connected units may be considered as a stainless steel mixer, a bronze dipping rack, and a steel conveyor cable. The iron tanks are so connected as to enable a transfer of the electrolytes between operations. Direction of current flow would depend not only upon the materials used, but also upon the concentration and types of electrolytes. However, a possible reaction could be that as indicated by the arrows. It is easily seen how a complex set of corrosion cells could develop, not only being destructive to the moving parts but to the tank as well.

An example of the other two types of metal-ion concentration-cell corrosion defined in Part I is illustrated by *Fig.* 16, where batch-mixing process equipment is represented with mixers, tanks, and all moving parts of steel. Apparently, this mono-metal construction would eliminate any corrosion difficulty of this type, and this would be true if the liquids remained at the same composition and concentration. However, as soon as the liquids in the vessels differ in these characteristics, there develops the potential for a corrosive action. A possible path of current flow is indicated by the arrows, assuming that the liquid in the middle tank differs in concentration or composition from the other two.

The corrective to eliminate this type of metal-ion concentration-cell action is the same as that prescribed for galvanic corrosion, i.e., insulation. As illustrated, this could be done by breaking the electrical connection between the tanks. For example, with *Fig.* 15 this could be obtained by insulating the valve flanges from the piping, as indicated by *Fig.* 12. This type is not to be confused with galvanic corrosion, which always involves two different metals in the same electrolyte.

Potentials Created at Part Junctions

Metal-to-metal junctions cause much trouble because of the oxygen-concentration-cell type of corrosion. Differences in oxygen concentration in the liquid layer in crevices between metals, as compared with that in the body of the liquid, produce this destructive action. This is also true with metal and nonmetal contact. Combinations of metal and wet porous material, such as cloth, paper, leather, etc., are not desirable because of this action. Pitting is also promoted by this type of corrosion.

With mildly corrosive conditions, such as outdoor exposure, coating the surfaces to be contacted with paint should be satisfactory. In severe cases, a plastic caulking compound should be used between the primed surfaces. It is recommended that these, in addition to being water repellent, be initially soft enough to conform to the metallic surfaces.

An example of the oxygen-concentration-cell type of corrosive action is indicated by *Fig.* 17. This steel valve stop plug was removed from a high-pressure

Fig. 18—Suction arrangement for hydraulic press pump which caused corrosion of valve stop plug shown in Fig. 17

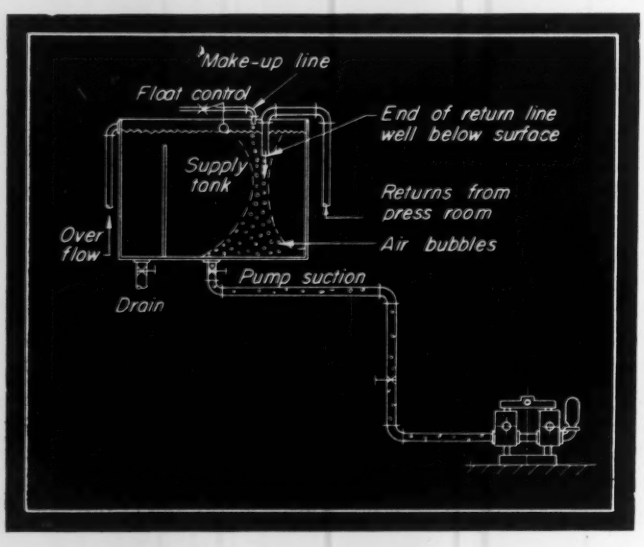
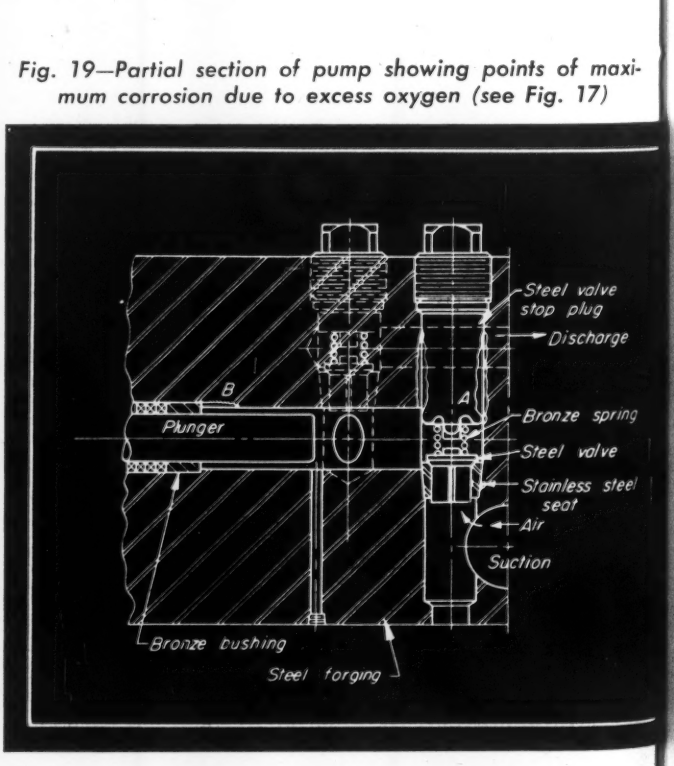


Fig. 19—Partial section of pump showing points of maximum corrosion due to excess oxygen (see Fig. 17)



pump, which was an integral part of a hydraulic system, *Fig.* 18. Entrained air (which concentrated the amount of oxygen that was already dissolved in the system) was drawn into the suction of the high-pressure pump. This produced accelerated corrosion of the oxygen-concentration-cell type at regions *A* and *B*, as indicated by sectional drawing *Fig.* 19. To help co-ordinate theory and design for a practical application the individual influences which produced this failure will be discussed.

Entrained air entered the suction valve below point *A* and was able to locate in the clearance (approximately 1/32-inch) between the valve stop plug and body. This produced an oxygen concentration-cell action, which increased the clearances, enabling more air (oxygen) to be entrained. The enlarged air pocket then produced a surging action with each plunger stroke, tending to remove the corrosive products (rust). This prevented it from acting as a protective coating; therefore, fresh metal was continuously exposed for attack.

It is to be noted that corrosion was severe at the point of contact of the bronze spring and steel valve stop. Galvanic corrosion could have started a cavity at this location, which could in turn produce an oxygen-concentration-cell action. Region *B* was also apparently attacked by this concentration-cell action and there would also be a possibility of galvanic action between the body and bronze bushing. However, as the attack is located only at the top of the cylinder, it would indicate that galvanic corrosion would be a minor factor.

A design to help combat this type of corrosion is illustrated by *Fig.* 20. As stainless alloy is highly successful in combating this type of corrosion, the valve stop is made into two parts for economic reasons as indicated. The valve-stop seating shoulder that was just below the base of the threads has been lowered toward the plunger cylinder, and now seats the stainless steel section of the valve stop. The sealing gasket prevents oxygen from becoming trapped farther up between the body and the steel section of the valve stop. The bronze spring is replaced by one of stainless steel, and the bronze bushing (not indicated in the illustration), can be replaced by an extended one, also of stainless steel.

It can readily be seen that the oxygen-concentration-cell type of corrosion may create serious problems. It is to be remembered by the designer, especially in cases such as these, that corrosion prevention does not necessarily start with the design of a specific machine. The designed system within which the machine is to be used as an integral part, should also be considered. Due to improperly designed baffles, the system as illustrated by *Fig.* 18 enabled considerable air to be entrained in the suction line of the high-pressure pump. With correctly designed baffling, much of the troublesome oxygen that was drawn into this pump could have been eliminated. As illustrated by *Fig.* 21, had they been so arranged by correct design to allow time for the air bubbles to rise to the surface and be expelled, much of the trouble could have been avoided.

Corrosion by pitting results from a highly localized corrosive action, causing rapid penetration of the metal. It usually is a combination of the examples described in the foregoing, and it's prevention would include the suggestions thus far offered.

If it is necessary for a rotating shaft to contact an enclosed corrosive solution, a definite problem is faced by the machine designer. For this, the stuffing box assembly, as illustrated by *Fig.* 11, is most commonly used. With a correctly designed assembly of this type, the independent sealing connection plays an important part. The principle is simple, yet important. The sealing fluid, usually clear water, is injected at a pressure slightly in excess of that produced by the enclosed corrosive solution. This tends not only to lubricate the packing, but to prevent the corrosive liquid from leaking out along the shaft. If it is correctly applied, only a slight leakage (approximately 60 drops per minute), of the sealing fluid results.

This type of assembly should not be designed with a shaft sleeve if at all possible because it is difficult to keep the corrosive solution sealed in such a way as not to leak between the sleeve and shaft. The question of sleeve design is primarily an economic one. Would it be cheaper to replace the shaft or the sleeve? With a small-diameter shaft, especially if it can be obtained from stock disbursement, it is best

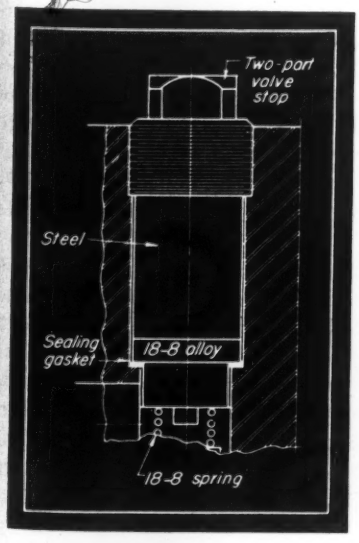


Fig. 20—Left — Correct materials selection plus redesign proved effective in eliminating corrosion of valve stop plug shown in Fig. 19

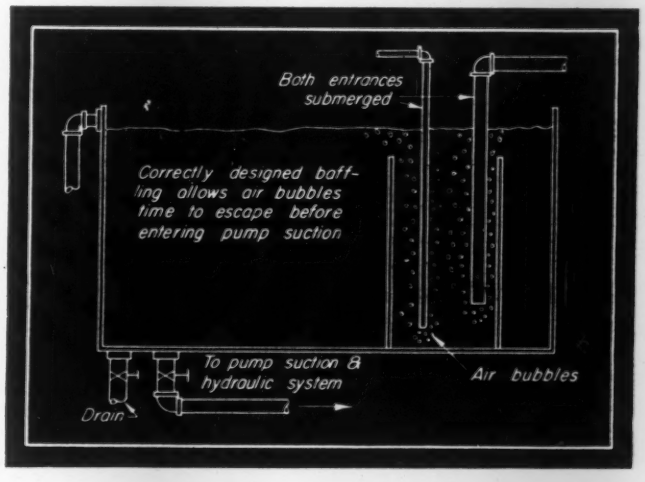

Both entrances submerged
Correctly designed baffling allows air bubbles time to escape before entering pump suction
To pump suction & hydraulic system
Air bubbles
Drain

Fig. 21—Right—Correctly designed baffling helps to eliminate oxygen concentration-cell corrosion in system of Fig. 18

to ignore the sleeve in the design.

If a sleeve is necessary, certain precautions should be taken, pertaining primarily to leakage. *Fig.* 22 illustrates a section of a mixer for an abrasive, corrosive solution, necessitating a shaft sleeve. In this case the mixing blade is fastened by a double sealing and locking nut, instead of the setscrew as shown in *Fig.* 11. The blade hub and sleeve are designed with different tapers so as to form a sealing connection. This is to prevent leakage under the sleeve. The design of the slinger makes it possible for a "chemical putty" to be used to prevent leakage from the box entering the joint.

Use of the mechanical seal for this type of service to eliminate stuffing-box difficulties for mild corrosive services is now being incorporated by some designers. This usually includes a stationary surface in conjunction with a rotating surface, forming a

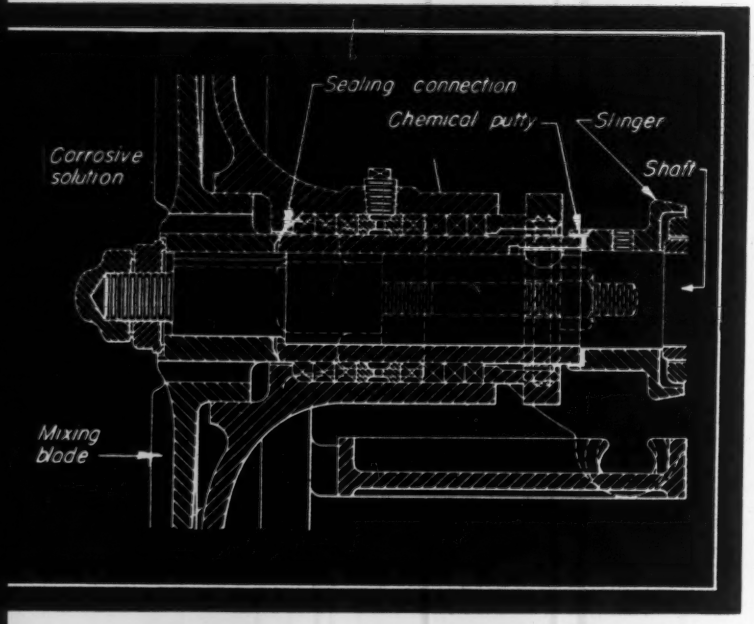


Fig. 22—Section through shaft-sealing stuffing box where shaft sleeve must be employed. At blade end, shaft-sleeve sealing is effected by metal-to-metal contact

sealing contact. The binding pressure for these sealing surfaces is supplied by springs or bellows and can be effected by external or internal assemblies. The former is the better design, considering corrosion, but it has definite pressure limitations. The latter, conversely, is not good for extreme chemical services, but for a system involving high pressures, has its advantages. The stuffing-box arrangement is used in the majority of cases, because of its simplicity.

Any type of machine which will be subjected to corrosive conditions, should be so designed as to facilitate draining. This is especially important in considering machine bases and supports. Periodic cleaning lengthens the service life of machines in many corrosive services or environments. This may not be a complete solution to all corrosion problems, but ease of periodic cleaning should be taken into consideration during design. This could be done by eliminating, if at all possible, short-radius fillets, sharp corners, overhangs and other such details.

If stainless steel is used, the machine designer should take into consideration the factor of heat treatment. For a stainless steel to give its full value of corrosion resistance, it should be correctly heat treated, i.e., the carbon should be in solution. If this condition does not exist, the carbon is present in the form of chromium carbides which may precipitate along the grain boundaries of the metal. Intergranular corrosion (Part I) may cause failure of machine parts constructed of an alloy so affected.

Even if the metal has been properly heat-treated, subsequent reheating to a sufficient temperature to put the carbon in solution followed by slow cooling will destroy the heat treatment effects. This is what happens with welding in localized sections. If welding is necessary, the design should be such as to make it possible for the parts to be reheat-treated. This cannot be done after the parts have been machined, because of warpage. Consequently, the sequence of operations should be: Welding first, reheat-treatment second and machining last. If this procedure is not feasible in the design, a careful welding technique should be specified, generally involving rapid welding with as little heat applied as necessary for the operation. Stainless steels to which stabilizers, i.e., columbium, have been added, are not affected in this manner. It should also be remembered that repair parts will be necessary. Consequently, the design should take into consideration ease of replacement of these expendable items. Corrosion resistant, alloyed valve liners are a good example of this.

It is apparent that materials of construction are important factors in the design of machines for corrosive service. Not only should they be considered as means to combat the corrosion types which have been discussed, but their effect on the final product may have to be considered. For example, machines in contact with drugs, foods, or chemicals must not change the odor, affect the flavor, or make the final product toxic. In some instances a catalytic effect, promoting the composition, may be involved.

Inasmuch as corrosion makes replacements necessary, no matter how careful the design, it is important that the material for a machine-part replacement be easily and quickly obtained. Even if a special alloy is better than a standard type for a certain corrosive condition, in many cases it might be better to employ the easily obtained, standard type in the design. This is an important consideration, especially for those machines which are directly related to plant production.

Combating corrosion in machines effectively requires the closest co-operation between designer and corrosion engineer. With such co-ordinated effort, definite steps can be taken toward curtailment of the tremendous losses incurred through corrosion. It is hoped that the information presented in these articles will help the designer to better understand the nature of corrosion and prompt him to adopt design measures such as can at least modify corrosion's destructive effects.

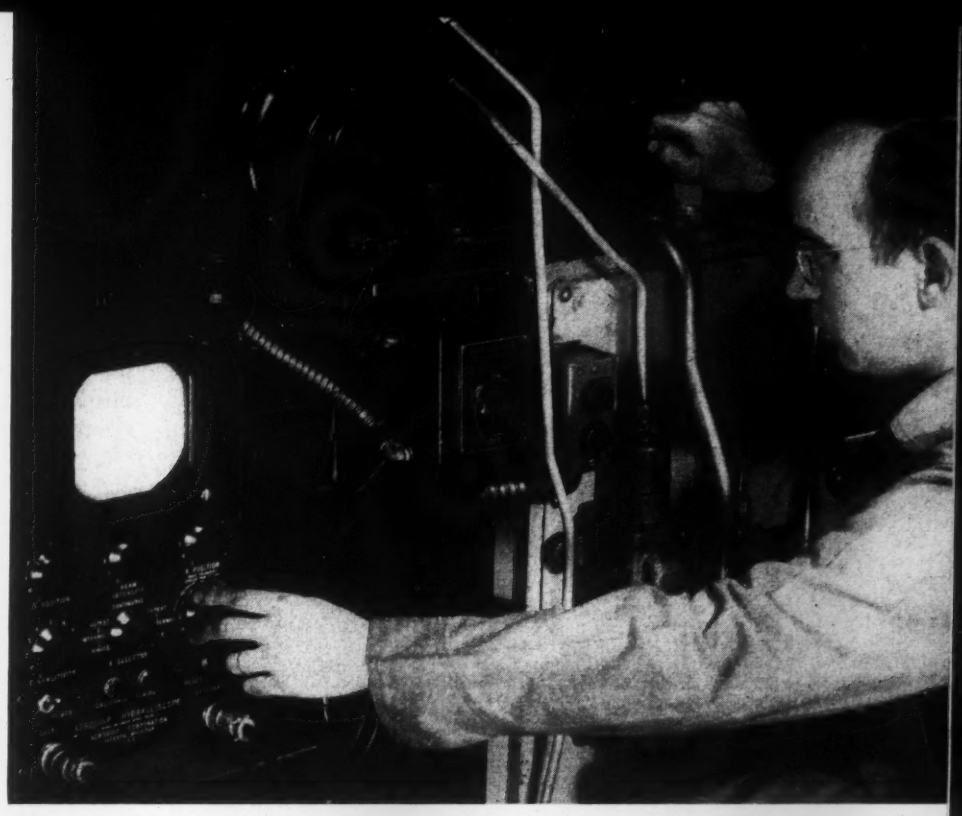


Fig. 1—Electronic analyzer establishes precise dynamic pressure conditions existing in hydraulic equipment, in a form readily convertible to a graph

By H. M. Reese
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The Parker Appliance Co.
Cleveland

Hydraulic Circuit Analysis

. . . shows that impulse pressures may be critical design consideration

ANALYSIS of an apparently well-designed hydraulic system, which persisted in rupturing steel tubing circuit members, points up a rather unusual circuit phenomenon, involving high-velocity impulse pressures. In this investigation an electronic analyzer, *Fig.* 1, was utilized to study the characteristics of the hydraulic circuit and to determine the cause of failure.

The mechanism involved is a high-speed hydraulic ram, used for inserting steel pins in rubber bushings. Normal system pressure was generally below the pump pressure of 1200 psi, and pressure surges were thought not to exceed peaks of approximately twice that figure. Nevertheless, 1-inch OD by 0.065-inch wall steel tubing having a bursting strength of approximately 8000 psi frequently failed, indicating undetermined system difficulties.

Details of the hydraulic circuit are shown schematically in *Fig.* 2. Hydraulic pressure for operating the ram is produced by a 1200 psi motor-driven hydraulic pump, supplemented by an accumulator pre-

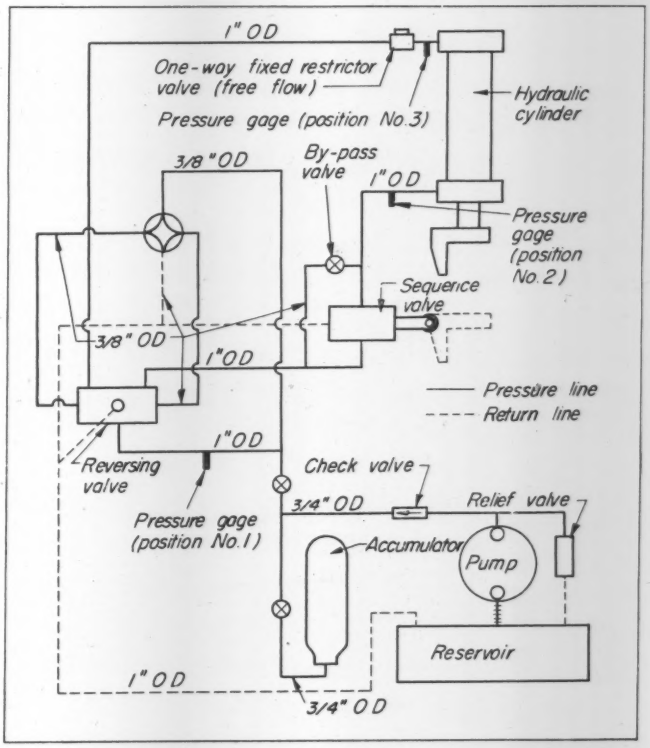


Fig. 2—Right—Schematic hydraulic circuit showing the three positions of the pressure gage for studying circuit pressure and flow characteristics in the system

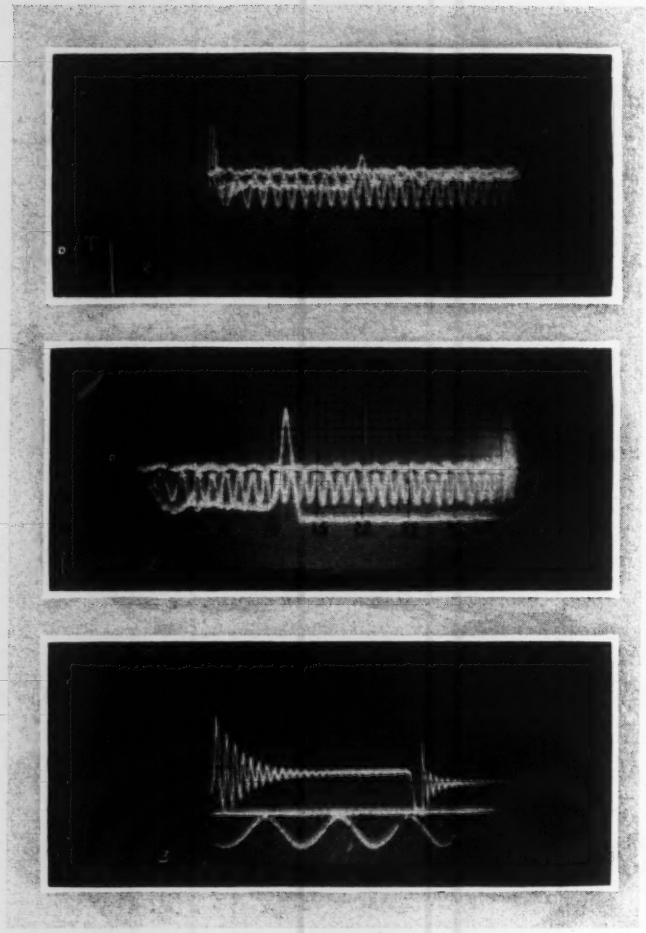


Fig. 3—Photographs of oscilloscope patterns, for the three positions tested, from which the analytical pressure charts were prepared. A reference line establishes the pressure level, while the variable-frequency sine wave pattern establishes time on the chart

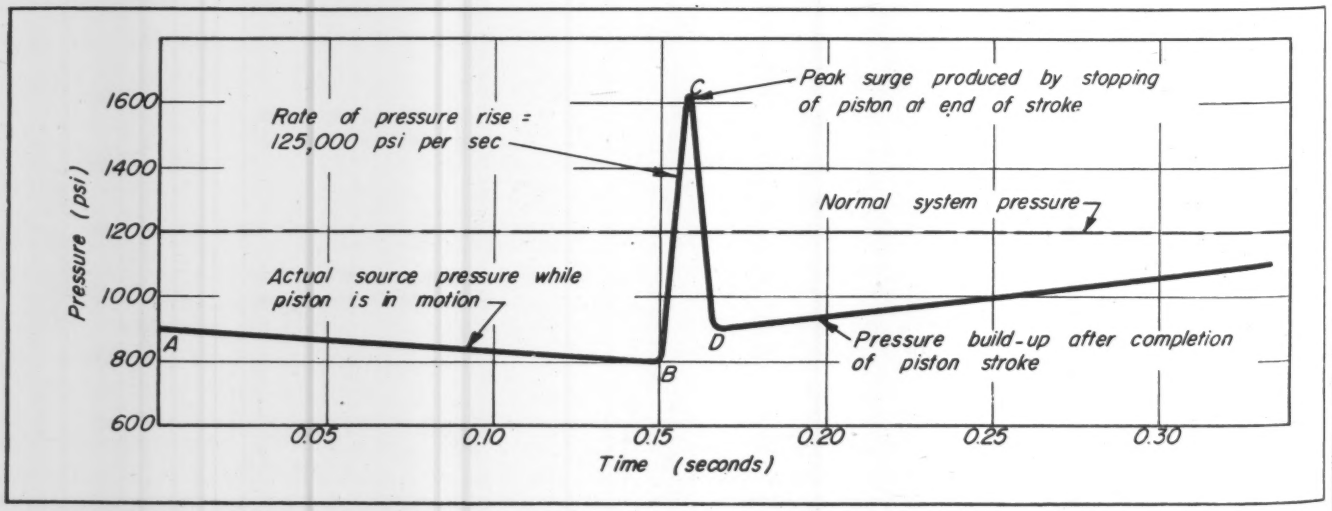
charged to 1200 psi. The accumulator is not of the conventional diaphragm type, but consists of a nitrogen bottle inverted so that the outlet is down, charged with nitrogen as needed. Thus the nitrogen in the bottle is in direct contact with the hydraulic oil, which of course has settled to the bottom of the bottle, covering the outlet port. The pump and accumulator pressure is conducted to a pilot-operated reversing valve through the 1-inch tube. The reversing valve is controlled by means of two hydraulic pilot valves which are actuated manually by the ram operator.

Valving for Ram Operation

When the reversing valve is positioned so as to operate the ram, i.e., to start downward movement of the piston in the cylinder, pressure is introduced into the cylinder through the upper port after first passing through a fixed one-way restrictor valve in the "free-flow" position. This pressure imparts motion to the piston in the cylinder, producing the desired ramming operation. Hydraulic oil on the connecting rod side of the piston is forced out of the lower cylinder port and returned to the reservoir through a sequence valve and a small by-pass valve which by-passes the sequence valve. When the piston approaches the end of its stroke downward, it trips the sequence valve, which is normally open, closing off the return line except for the small return opening provided by the almost-closed by-pass valve. This action traps a small amount of hydraulic oil in the bottom of the cylinder and line between the cylinder and the sequence valve, providing a cushioned hydraulic stop for the piston in its downward motion. The pressure produced in stopping the piston is then relieved relatively slowly through the by-pass valve, eliminating any piston bounceback.

When the reversing valve is reversed to return the piston to its upper position, hydraulic oil pressure is introduced into the lower cylinder port, first through the small by-pass line, and then, after the sequence valve is again opened by the return motion of the piston, directly through the sequence valve and the

Fig. 4—Below—Plot of hydraulic oil pressure vs. time in one-inch line connecting pump and accumulator to reversing valve. Pressure pick-up is at Position No. 1 as shown in the schematic hydraulic circuit in Fig. 2



by-pass circuit. The velocity of the return stroke of the piston is kept low by maintaining back pressure in the upper portion of the cylinder by means of the one-way fixed restrictor valve in the line between the upper cylinder port and the reversing valve. The sizes of hydraulic lines used are indicated in *Fig.* 2; all valves are of sizes equivalent to the connecting lines.

To pinpoint the troublemaking factors in the circuit, the electronic pressure analyzer (*Fig.* 1) especially designed for hydraulic work was employed. The device consists of a super-sensitive pressure pick-up which can be inserted in a circuit and connected to a cathode-ray oscilloscope with photographic attachment to obtain in minute detail the actual pressure conditions existing in the circuit being examined. A recording vibrometer was also used to measure the magnitude and frequency of vibration introduced into the system.

Pressure Tested at Three Positions

The hydraulic pressure pick-up was first installed in the line connecting the pump and accumulator to the pressure port of the reversing valve, position No. 1 in *Fig.* 2. The electrical connection between the pressure pick-up and the analyzer was then made. A precision switch was clamped to the ram in such a position as to make contact and energize the analyzer's tracing beam at the instant the piston was set in motion. The ram was then operated and photographs taken of the pressure pattern during the down-stroke of the piston.

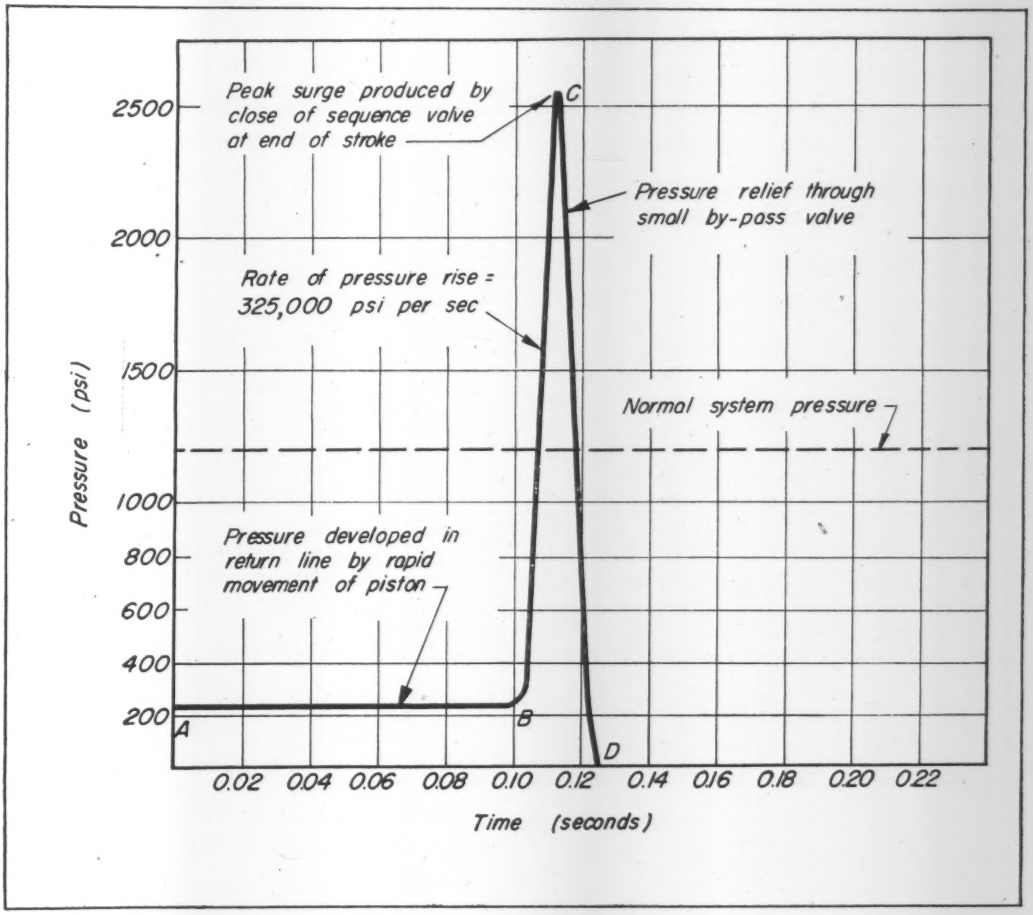
The pick-up was next installed in the line connecting the lower cylinder port and the sequence valve as shown in position No. 2, *Fig.* 2. With the switch in the same position, the ram was again operated and photographs taken at this position in the hydraulic circuit.

The pressure pick-up was finally installed in the line between the upper cylinder port and the one-way fixed restrictor valve, Position No. 3. Photographs were similarly taken at this position with the ram in operation.

The photos obtained, *Fig.* 3, show the pressure phenomena in relation to time at the three different locations in the hydraulic circuit of the ram during the working stroke. Curves were constructed from this data to scale for study. An analysis of the respective curves follows:

POSITION NO. 1: Portion of the curve between *A* and *B* in *Fig.* 4 represents the source pressure as supplied by the pumps and accumulator while the piston is in motion in its downward direction. The gradual drop-off in pressure results from the high velocity of the moving piston. At point *B*, the sequence valve near the end of the piston stroke closes, trapping the fluid being exhausted from the cylinder by the motion of the piston. The resulting fast stop-

Fig. 5—Pressure-time curve for conditions in one-inch line connecting lower cylinder part to sequence valve. Pick-up is at Position No. 2



ping of the piston produces a rapid build-up of pump and accumulator pressure to a peak surge of 1620 psi, as illustrated by the portion of the curve from *B* to *C*. The drop-off of this peak surge pressure, as represented by the portion of the curve between *C* and *D*, results from by-passing the trapped fluid at the bottom of the cylinder. The portion of the curve from *D* on, represents the gradual pressure build-up of the accumulator by the pump after completion of the piston stroke.

POSITION NO. 2: Portion of the curve between *A* and *B* in *Fig.* 5 represents the exhaust pressure of 225 psi developed in the return line by the rapid movement of the piston in the cylinder. At point *B*, the sequence valve closes, producing the fluid trap at the bottom of the cylinder and stopping the downward movement of the piston. The rate of pressure rise of this trapped fluid is represented by the portion of the curve between *B* and *C*, producing a peak surge pressure in the return line of 2550 psi. This peak surge pressure is relieved by exhausting the trapped fluid through the small by-pass valve, as illustrated by the steep pressure-drop portion of the curve between *C* and *D*.

POSITION NO. 3: The portion of the curve between *A* and *B* in *Fig.* 6 represents the pressure sequence developed at the upper cylinder port when the rapidly moving piston is suddenly stopped. The fluid pressure, which has produced the movement of the piston, suddenly builds up when the piston is stopped, reaching a peak surge pressure of 2400 psi. In a period of approximately 0.3-second, this surge pressure levels off to 900 psi as represented by the portion of the curve between *B* and *C*. At point *C*, the reversing valve is actuated causing the pressure to drop to zero as represented by portion of curve between *C* and *D*. By reversal, pressure is introduced into the lower cylinder port causing return motion of the piston. Upward motion of the piston causes exhausting of fluid in the cylinder out of the upper cylinder port which is restricted by means of the restrictor valve adjacent to this port. The pressure situation developed by this action is illustrated by the portion of the curve between *D* and *E*. The peak surge pressure is found to be 1800 psi.

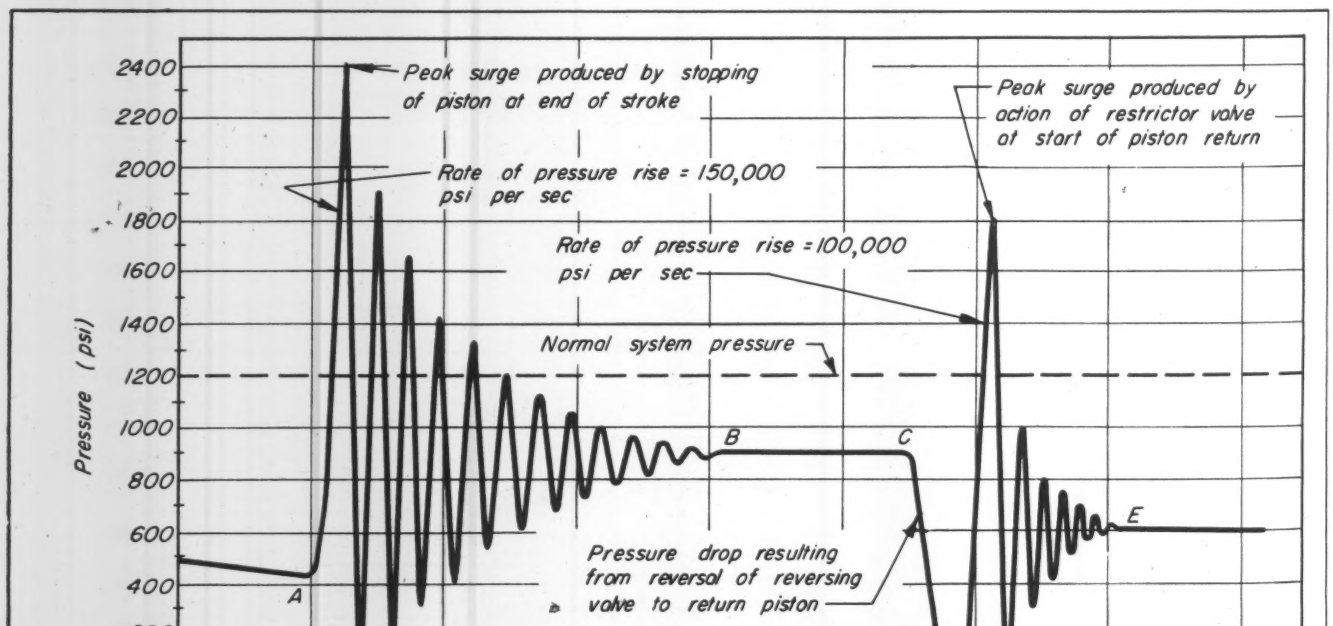
Steep Slopes Indicate Shock Loads

These graphs show conclusively that the pressures occurring in the system at no point exceed those anticipated. The highest reached is but 2550 psi, giving a safety factor of 3 in terms of the strength of the tubing which continually fails.

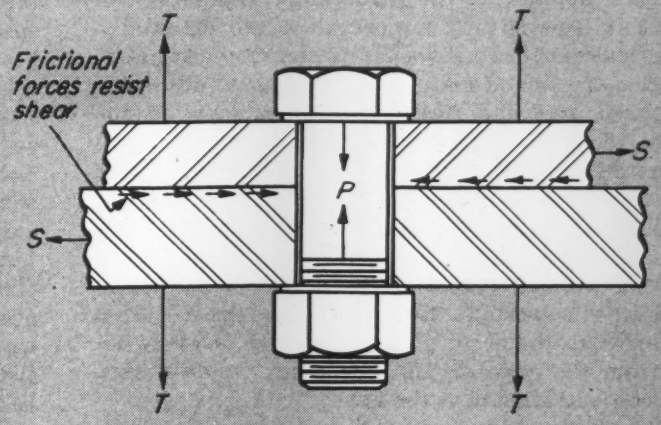
The steep slopes of the curves where pressure rise occurs, however, give an immediate answer to the failure problem. In *Fig.* 5, for example, the surge peak of 2550 psi is reached in 0.01-second, and the rate of build-up is 325,000 psi per second. Thus the velocity of application of an inherently modest pressure subjects the affected portion of the system to a most severe shock load.

The direct remedy for such high velocity impulse pressures was applied in this case, and heavier tubing with double-weight, triple-type fittings was installed to raise the "safety factor"—in terms of the peak pressure reached—to 4.

Fig. 6—Position No. 3 for pressure pick up gives this pressure-time curve at upper cylinder port



Bolted Assemblies



Factors to be considered in specifying and applying threaded fastenings. Discussion includes material selection and proper amount of torquing

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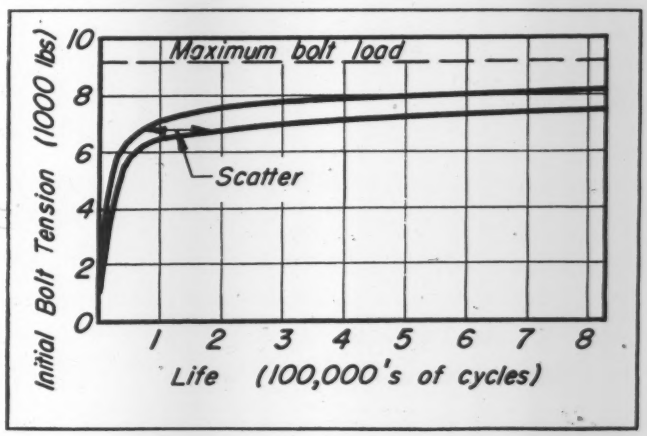
SO UNIVERSAL is the dependence on bolts and similar fastening devices for the assembly of every conceivable kind of machine, that periodic reappraisal of their characteristics is imperative if maximum effectiveness is to be attained at lowest cost. Such a reappraisal is presented in this article, which also includes discussion of some recent and current developments leading toward a better understanding of how to specify and apply bolts.

Bolts are engineered products capable of satisfactory control. Their uses, however, are practically infinite over a wide range of service conditions which modify and determine their behavior. Most of the difficulties in the use of bolts stem from the problems of producing preloading during tightening and maintaining it throughout the life of the bolted device.

The function of preloading is indicated in *Fig.* 1, in which preloading tension of the bolt is indicated as *P*, the tension developed in the bolt shank by torquing the bolt head or nut, or by other means. Tensile forces, *T*, may be applied to the bolted structure, but so long as their reaction at the bolt does not exceed *P*, there is no separation of the parts nor extension of the bolt. Therefore, there is no increase in the tensile load on the bolt. In other words, the tension in the bolt remains at the preload value, *P*. This is not strictly true, but is for all practical purposes if the bolted material is well fitted and is rigid compared with the bolt. If the bolted material is not comparatively rigid, but is elastic, such as by the in-

Fig. 1—Top—Bolt preload, P, resists tensile forces, T, direct and shear forces, S, by virtue of friction

Fig. 2—Below—How preloading, an essential factor in fatigue resistance, increases endurance life of a bolt



Based on a paper presented at the recent annual meeting of the SAE in Detroit.

terposition of a gasket material, then the bolt load will be increased beyond the preload value as tensile forces are applied to the structure. However, the added bolt load is only a portion of the externally applied load and becomes a small and negligible portion when the bolted material is comparatively rigid.

Similarly, shearing forces, *S, Fig.* 1, may be applied to the structure, but the joint does not slip if the frictional resistance between the parts as produced by the preload tension, *P,* is not exceeded by the shearing forces. Since the joint does not slip, there is no shear on the bolt nor change in load from its initial preload value. Such a connection offers the further advantage that stress in one plate is transferred to the other plate by friction at areas remote from the edge of the hole, thus avoiding high stress concentration near the hole.

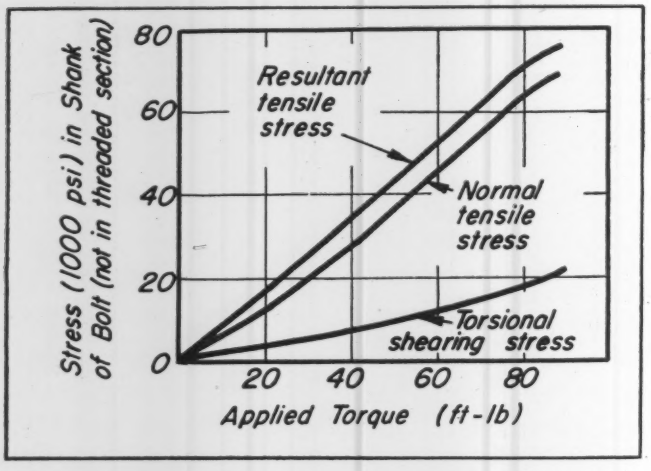


Fig. 3—Torquing the nut adds a torsional shearing stress in bolt shank, producing a high resultant stress

Some typical uses of preloading of bolts follow. Carr[1] has pointed out, in connection with flange bolts resisting gas or fluid pressure, the necessity of tightening the bolts to a point sufficient to resist the operating forces and to maintain gasket compression which prevents leakage. Preloading of bolts is also useful in applications where the joint generally is loaded in a shearing direction. Almen[2] has shown in applications where loading is in a tensile direction that preloading of bolts is an essential factor in resistance to fatigue. His experiments, *Fig.* 2, indicate that the endurance life of a bolt in comparatively rigid parts increases rapidly as the bolt is progressively tightened and becomes infinite as the initial bolt tension approaches the maximum bolt load applied to the assembly. The reason for this is that increased preloading reduces the amplitude of stress change in the bolt to a safe value, provided the bolted material is comparatively rigid. This resistance of the bolt to fatigue is dependent solely upon the preloading of the bolt and is not affected by factors such as surface conditions of the bolt or similar stress-raisers that might otherwise lead to early bolt failure.

However, if the bolted material became progressively less rigid, there would be a stress cycle of increasing amplitude in the bolt, which might lead to early failure. In such a condition, it is necessary to pay special attention to proper filleting under the bolt head, surface finish, contour and finish of thread roots, and similar factors that reduce stress concentration. Rounding of the thread roots is beneficial, as is also the rolling and cold forming of threads. Rolling of threads after heat treatment is done in some special cases and gives the most beneficial results. Rolling provides a smoother finish of the thread roots, increases the tensile strength of material at the root by cold work, and sets up compressive stresses in the material at the root, all of which increase the endurance limit of the bolt to nearly twice its normal value. Heat treatment subsequent to thread rolling, of course, nullifies the cold working and eliminates the compressive stresses, but even so, such threads usually are superior because of their better finish.

Preloading is also beneficial in providing a locking effect to avoid unscrewing. Goodier and Sweeney[3] demonstrated that when there is an excessive amplitude of stress change in a bolt, there is a tendency for the nut to loosen. Proper preloading of bolts and proper design of bolted parts reduce the amplitude of stress change and thereby provide a certain locking effect. Of course, it is not always possible, because of design requirements, to provide rigid abutments, and even though a bolt is preloaded, there still may be a considerable amplitude of stress change. In such cases locking may be provided by any one of the many types of locking devices available.

It is seen, therefore, that preloading of bolts is an essential factor in almost all important applications. There are casual applications of bolts where con-

TABLE I

Torque Coefficients

Bolt Size	Torque coefficients—R—		
	*Theoretical	Measured (average) High-point torque	Mid-point torque
¼ -20	0.210	0.243	0.267
¼ -28	0.205	0.216	0.231
5/16 -18	0.210	0.206	0.186
5/16 -24	0.205	0.194	0.183
⅜ -16	0.204	0.200	0.247
⅜ -24	0.198	0.192	0.234
7/16 -14	0.205	0.217	0.224
7/16 -20	0.200	0.194	0.190
½ -13	0.201	0.205	0.158
½ -20	0.195	0.167	0.205
9/16 -12	0.198	0.194	0.214
9/16 -18	0.193	0.196	0.207
⅝ -11	0.199	0.178	0.196
⅝ -18	0.193	0.183	0.175
¾ -10	0.194	0.169	0.172
¾ -16	0.189	0.170	0.180
⅞ - 9	0.194	0.181	0.194
⅞ -14	0.189	0.171	0.178
1- 8	0.193	0.188	0.204
1-14	0.188	0.161	0.167
Average	0.198	0.191	0.201

*Computed with coefficient of friction of 0.15 and the dimensions of American Standard Regular Nuts.

[1] References are tabulated at end of article.

trolled preloading may not be necessary, but this is not true of most applications in important machines and structures.

Preloading of bolts may be accomplished in several ways. Very large bolts often are drilled along the axis for the insertion of a heating element. The bolts are heated to a predetermined temperature and lightly torqued in place. Subsequent thermal contraction of the bolt induces preloading, the amount of which is dependent upon the temperature of heating. A hydraulic jack arrangement has been provided in some cases to stretch large bolts and permit the nut to be assembled easily. On release of the jack, its load is transferred to the bolt. Power devices also are available for tightening. The method most generally used is the ordinary hand wrench, often equipped with a torque-measuring device.

Where precise preloading of bolts is required, it is usual to determine this preloading by measurement of the elastic extension of the bolt as the bolt or nut is torqued. By calibration, the required elastic extension of the bolt for a given load is determined. This procedure has obvious disadvantages for high production work. The more usual method is to measure the amount of torque applied to the bolt head or nut and estimate from this what the preloading of the bolt is.

Friction Absorbs Most of Torque

The relationship between applied torque and induced tension is not fixed. Ninety per cent or more of the applied torque is consumed in friction on the bearing face of the bolt head or nut and on the bearing face of mating threads, and produces no tension whatever. The degree of friction is highly variable and it is apparent that the relationship between torque and tension will be almost proportionately variable, since most of the torque is consumed by friction. In recent investigations Maney[4] tested a large number of bolts to determine the variation of the torque-tension ratio. In these tests bolts and nuts were assembled between the heads of a regular tension testing machine in such a way that torquing of the bolt head or nut would load the machine and the load would be indicated on the machine dial. The bolts used in this investigation had cold-formed threads and were of low carbon steel, stress relieved. The nuts were of American Standard regular proportions, semifinished. A quantity of bolts and nuts from each of several manufacturers in each size and thread pitch were tested, with the average results shown in TABLE I.

From theoretical considerations it is known that the torque required to produce a given tension in the bolt is equal to the torque coefficient, R, times the nominal bolt diameter times the bolt tension. If the torque is given in inch-pounds, the bolt diameter in inches, and the bolt tension in pounds, the torque coefficient is about 0.2. Its computed value is shown in TABLE I, together with the average measured values taken at a high-point torque and also at a mid-point torque. It is apparent from these data that the torque coefficient is approximately a constant for a given set of friction conditions and is nearly the same for all diameters and for both coarse and fine threads.

Data shown in TABLE I are average. In a group of, say, twenty-five bolts, it may be found that an extreme bolt may depart from the average by some thirty per cent. The average deviation, however, is about seven per cent. It should be understood that these data apply only to bolts and nuts in the stated condition. The torque coefficient will vary even more with variations of surface conditions such as plating and lubrication.

The question naturally arises as to how much preload should be applied to a bolt. Maney[5] has shown that when a nut is torqued on a bolt there is not only a normal tensile stress induced in the bolt, but a torsional shear stress as well. This torsional shear stress results from the frictional drag between the threads as the nut is turned, which consumes about forty-five per cent of the applied torque. *Fig.* 3 shows the results of measurement of these stresses by electrical strain gages attached to the bolt shank. Bolts used were of 1020 steel, stress-relieved. The normal tensile stress is the stress providing clamping force or preloading of the bolt. Because of the concurrent torsional shearing stress, the resultant tensile stress in the bolt is higher than the normal stress. This means that if the bolt were torqued to failure it would break at a lower load than would cause failure under direct tension. It is probable that a bolt of this grade should not be depended upon to produce a clamping load beyond about two-thirds of its tensile strength. However, torquing until the bolt just begins to yield should not cause damage, provided massive yielding does not occur which would cause considerable distortion of the fastener.

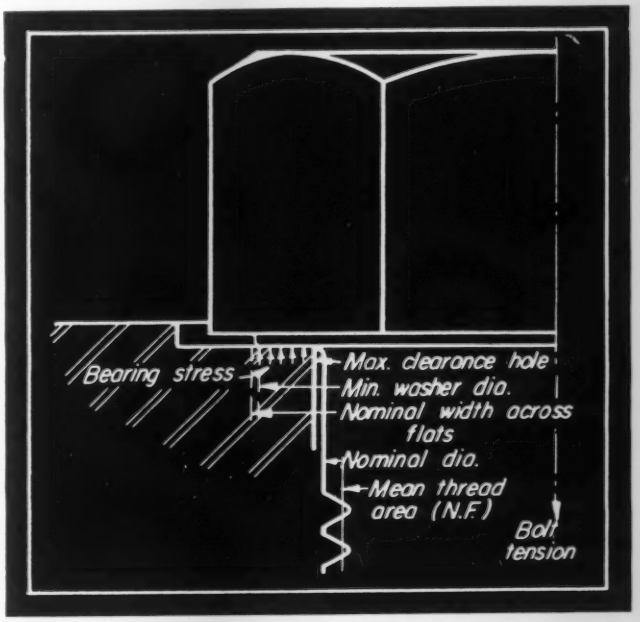


Fig. 4—High bearing stress under bolt head or nut can cause relaxation of initial tension due to crushing

While a bolt fails under torque at less than its

direct tensile strength, Hill[6] has shown that the ultimate strength has not been reduced by torquing if torquing is not continued to failure. In other words, if a bolt were torqued to a point less than the torque failure load, then the subsequent load in direct tension to break it is the same as it would have been were the bolt pulled without torquing.

There are special conditions where the preload of the bolt should be lower than that previously suggested. If the bolted parts are flexible relative to the bolt, then externally applied loads will increase the bolt stress beyond the preload value by an amount depending upon the relative elasticities. In these applications the preload should be at a low enough point so that the additional stress will not cause yielding of the bolts.

In view of the importance of preloading, attention should be given to the maintenance of the preloading throughout the life of the equipment in which the bolt is assembled. It is probable that the bolt tension relaxes to some extent in service. Relaxation may be due to flattening out of rough spots on contact surfaces or to extrusion of dirt, paint, or plating from contact surfaces. There also may be a flow of metal at local points of high stress concentration.

TABLE II

Approximate Grades of Bolts

Type	Tensile Strength	Application
Common	50,000	Machine, Carriage, Lag, Plow Bolts, etc.
Low-Carbon, Cold Headed, Stress Relieved	80,000	Automotive Bolts
High-Carbon, Quenched & Tempered	105,000	Automotive Bolts
Alloy, Quenched & Tempered	125,000	Aircraft Bolts & Studs

Relaxation of initial tension might also occur in service either by crushing of the bolted material under the bearing face of the bolt head or nut or by crushing of bearing faces of the bolt or nut. The degree of such crushing will depend upon the net area of bearing face in contact, *Fig.* 4, and the relative physical properties of the bolt and bolted material. The bolt head or nut should be wide enough so that with the maximum clearance hole expected there is sufficient contact area so that the fastener material will not crush under medium bolt load. Even so, the bolted material may crush if hard fasteners are used in softer bolted parts, unless extra contact area is provided either by the fasteners themselves or by the addition of a washer harder than the bolted material.

As mating surfaces adjust themselves, relaxation reaches a stable value with bolt tension somewhat less than the initial preloading. For this reason, where preloading is of critical importance, a bolt of sufficient size should be used so that it can be preloaded to a value in excess of requirements, thereby providing for anticipated relaxation.

Loss of bolt grip due to relaxation is generally a fixed amount regardless of total grip length. The effect of relaxation, therefore, is more important in short bolts than it is in long bolts, since long bolts are assembled with a greater amount of compensating elastic elongation. Additional elastic elongation is often provided in bolts and studs by "waisting" or reduction of the unthreaded portion of the bolt. This reduction in section does not weaken the bolt below that of the threaded section, but results in higher tensile stresses in the shank and consequently greater elastic elongation. Another method of obtaining additional elongation is by the use of a patented flexible head. Further elongation is provided by elastic deformation of the threads themselves by bending and by elastic distortion of the nut. No investigation is known which reveals the amount of this elongation or of the relative amounts occurring in fine and coarse threads, but the amount probably is of comparable magnitude to that occurring in the body of the bolt.

Nut-Deformation Affects Thread Loading

Another element of the fastener which should be considered is the effect of the bolt load upon the nut. Goodier[7] has shown that the nut expands as a result of radial forces on it created by the bolt tension acting through the wedge angle of the thread. This dilation of the nut tends to be greater near the bottom of the nut because of the non-uniform distribution of load on successive threads within the nut. If the pitch of mating threads is accurate before loading, then when the load is applied, the bolt tends to stretch and increase its thread pitch, whereas the nut tends to compress and shorten its thread pitch. The result is that the threads near the base of the nut take more load than the top threads. Buckingham[8] calculated the distribution of load on successive threads, *Fig.* 5, for ½-13 and ½-20 threads. For coarse threads the loads vary from 53 per cent of the average at the top of the nut to 179 per cent at the base of the nut, while with fine threads the loads vary from 37 per cent of the average at the top of the nut to 231 per cent at the base.

In the foregoing, the calculation is based only on thread bending and neglects the radial dilation of the nut, which would tend to make the load distribution somewhat more uniform. In actual practice with relatively soft bolts and nuts the load distribution is not as bad as that illustrated, since the highly loaded threads may shift plastically and thereby relieve themselves of some of the load, transferring it to the less severely loaded threads. This plastic adjustment of threads cannot be accomplished so well if both the bolt and nut are of relatively hard material. For this reason, when high strength bolts are selected, it is well to use a nut of as soft a material as will fully load the bolt. This usually can be done, since the nut proportions are generally over-designed.

Bolts and nuts, of course, are made from a variety of materials and by a variety of processes. In some cases the exact chemical composition of the material is important, such as where corrosion is a factor. Also, there are occasions where composition and metallurgical treatment must be closely determined so as

to get a special service result that otherwise cannot be controlled readily. For example, "creep" of steel at high temperature is not indicated by short-time tests, since it takes a long time to measure this property adequately. In such cases control of composition and treatment is customary. However, in the ordinary usage of steel bolts and nuts, the exact chemical composition of the material and the process of manufacture are of little interest to the user, provided service requirements are met. In fact, it is a mistake to limit these production factors in specifications, since it reduces the flexibility of production control in the manufacturer's plant and results in unnecessarily high costs. What should be done is to specify the requirements in terms of useful service factors and leave material and manufacturing procedure to the producer.

Simplifying Bolt Grade Selection

Further, in order to simplify production, it would be desirable to select a standard physical grade of bolt rather than to pick at random a set of physical properties from the almost continuous series that it is possible to produce from the lowest strength to the highest strength bolt. Though grades of bolts are not fully standardized, an approximate gradation suitable for most applications is shown in TABLE II. These grades are classified in terms of tensile strength, because it is a property most generally understood. However, the easiest and most economical physical test for acceptance of bolts is a hardness test, which generally gives all the information required for control of bolt production. Where a tensile test is made it should be performed preferably on the full-size bolt rather than on a prepared specimen.

Within the grades of bolts outlined in TABLE II the common grade, of course, costs the least and is generally used for ordinary machine, carriage, lag, plow bolts, and such applications. Recently the American Society for Testing Materials has prepared a specification, A307-47T, covering machine bolts of this grade.

Bolt Grades for Higher Strength

The next grade, widely used in automotive practice as a hexagon-head bolt, is cold headed from low carbon steel and stress relieved to an extent that will prevent head failure, but not such as to soften the bolt completely. The resulting product, therefore, retains much of the strength benefit of the cold working. Where higher physical properties are required, and particularly where high preloading of the bolt is necessary, a quenched and tempered high-carbon steel bolt is most economical, since it provides possibly the highest clamping force per dollar of cost. It also results in other economies since its higher strength permits the use of smaller bolts and the attendant greater compactness of design of bolted parts.

Where still higher physical properties are required, it is generally necessary to use an alloy steel, particularly in larger bolts. Even here it is not usual to specify the exact alloy, since there is a wide range of alloys that will give equivalent hardenability and equivalent service results. While generally of high cost, these bolts are useful in applications such as aircraft construction, where maximum compactness of joint and saving of weight justifies the higher cost.

There are special applications where still higher physical properties than those shown in TABLE II are required. The use of such bolts, however, is special and requires considerable care in the manufacture of the bolt and in the preparation of the bolted material. Sachs[9] investigated the strength of bolts having a ten-degree beveled shim under the head, producing eccentric loading, *Fig.* 6. Comparisons were made between the ultimate strength of the bolt having a shim and of corresponding turned test specimens, both being heat treated to the same condition. The bolts and prepared specimens gave comparable results only when tempered to strengths up to about 150,000 psi. At higher specimen strengths the eccentrically loaded bolt suffered severe loss of strength. Also, Brunot and Schmittner[10] in a study of studs for a steam chest found superimposed bending

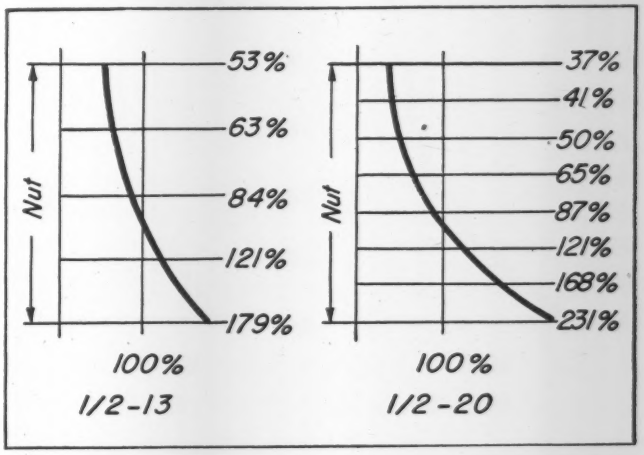


Fig. 5—Calculated stress distribution on threads of nut is nonuniform due to bolt stretch and nut compression

stresses in the studs of as much as 100 per cent of the direct tensile stress. This is the result of bending produced by inaccuracies in the dimensions of the fasteners and the bolted parts. The conclusion drawn from this information is that special care must be taken in the manufacture and use of very high strength bolts to avoid eccentric loading if this strength is to be realized in practice.

The principal function of the nut is to load a bolt, and selection of the grade of nut should depend upon the bolt to be used. Nuts may be classified by their stripping strength, which is determined by engaging the nut on a hardened and ground threaded mandrel which is pulled through the nut. The stress at failure is determined by dividing the load by the so-called mean thread area of the threaded section. Common nuts will withstand a strip test of this variety of about 110,000 psi and, therefore, are suitable for use with all of the carbon steel bolts listed in TABLE II. For alloy steel bolts and special bolts a nut of higher

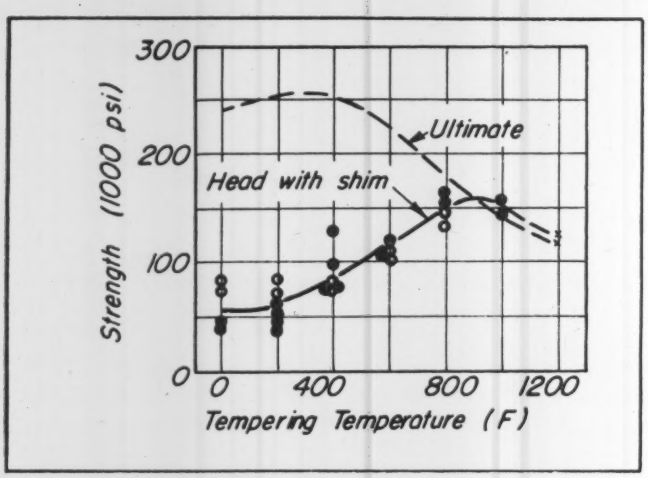


Fig. 6—Effect of eccentric loading, due to insertion of a beveled shim under the bolt head, on bolt strength

stripping strength is required, but its hardness should be kept as low as possible consistent with the required stripping strength. This permits the nut threads to conform to those of the harder bolt without excessively loading any thread.

It should be realized that a bolt is a specific commodity readily capable of satisfactory control. Its inherent properties can be known and are definite. The usefulness of these properies, however, is modified greatly by conditions of use. Improper design of bolted parts or improper assembly can destroy the utility of even the finest bolt. Since applications vary over such a wide range, it is difficult to establish standard rules for the use of bolts. What is needed is more knowledge of the necessary conditions of use, together with an investigation of how the bolt can best serve these uses. Many consumers of bolts are engaged in such a study in their own plants, often duplicating work already done or being done by others. It would be of economic advantage to all and would provide a most efficient procedure if one of the professional engineering societies in the mechanical field should sponsor a joint research group that would compile and authoritatively codify present knowledge and engage in research where important problems are found to exist.

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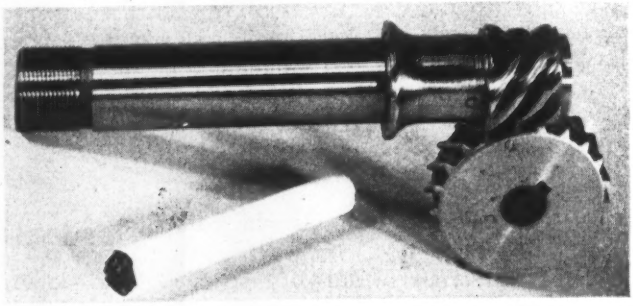
Miniature One-Horsepower Drive

WITH a center distance only 1 1/32 inches, the Cone drive illustrated transmits over one horsepower with an input pinion speed of 9000 rpm. It is employed on Douglas DC-6 aircraft to drive the oil pump for the cabin supercharger. At takeoff speed the pump delivers 7 gallons per minute of oil



at 155 psi, the corresponding torque being 30 lb-in. Compactness of the drive, which provides a 3 to 1 speed reduction to the pump, is well illustrated by the size comparison with an ordinary cigarette.

Hydraulic Clutch

CORRECTION: In the article "Hydraulic Clutch" which appeared in the January issue, the following corrections should be made to equations on Page 118:

Equation 14 should read:

$$\frac{N}{N_i} = \left[1 - \frac{CC_o t}{CA_o + C_s}\right]^n$$

$$\left\{ - \frac{T_f A_o}{N_i J C_o} \left(\frac{1}{1-n}\right) \left[\left(1 - \frac{CC_o t}{CA_o + C_s}\right)^{1-n} - 1\right] + 1 \right\} \quad (14)$$

where

$$n = \frac{D^2}{JC_oC}$$

Equation 15 should read:

$$\frac{N_l}{N_i} = 1 - \left[1 - \frac{CC_o t}{CA_o + C_s}\right]^n$$

$$\left\{ - \frac{T_f A_o}{N_i J C_o} \left(\frac{1}{1-n}\right) \left[\left(1 - \frac{CC_o t}{CA_o + C_s}\right)^{1-n} - 1\right] + 1 \right\} \quad (15)$$

and Equation 18 should read:

$$\frac{dN}{dt} = \frac{-nCC_oN_i}{CA_o + C_s} \left[1 - \frac{CC_o}{CA_o + C_s} t\right]^{n-1} \quad (18)$$

It is greatly regretted that these errors appeared in the article, and it is hoped that they caused no undue inconvenience to our readers.—ED.

MACHINE Editorial DESIGN

Technological Aid to Europe

Can engineers assist in the successful carrying out of the European recovery program? Of the numerous proposals put forward by leading industrialists, one plan in particular envisages the establishment of an industrial council consisting of top-ranking technical experts and administrators whose services could be loaned temporarily to Western European countries. Such assistance, it is claimed, would keep the actual cash loaned or turned over to the countries involved to a minimum, and would reduce the burden on the American taxpayer besides serving to supplement the various aspects of the European aid program.

Roy W. Gifford, chairman of the board of the Borg-Warner International Corporation, who advocates this procedure, feels that the primary aim of the industrial council would be to increase the productivity per man in European industries and companies requesting such assistance. The period spent abroad by consultants from the council would, of course, only be sufficiently long to modernize industrial plants and to assist in the adoption of improved methods and processes.

That the Western European nations could well afford to take advantage of assistance of this nature is immediately apparent from comparison of the technological advancement made in this country as compared to the slow rate of progress over there. In a recent study made by the Machinery and Allied Products Institute it is pointed out, for instance, that in Great Britain the output per worker in representative industries in prewar years was only one-third that of the output of the American worker. The study brings out that the major reasons for this lack of efficiency on the part of workers in Britain are the obsolete machinery and outmoded methods of production under which they operate.

It could be said that technological aid from the United States, as proposed, might eventually react to our detriment. There is no doubt, however, that national prosperity cannot long be maintained without a reasonable amount of worldwide prosperity. Further, the scientists, engineers and designers who have been so successful in bringing America to its leading position can without question be relied upon to continue their accomplishments and thus keep this country in the forefront with respect to the technological knowledge and advancement upon which our economic system is based.

L. E. Jerry

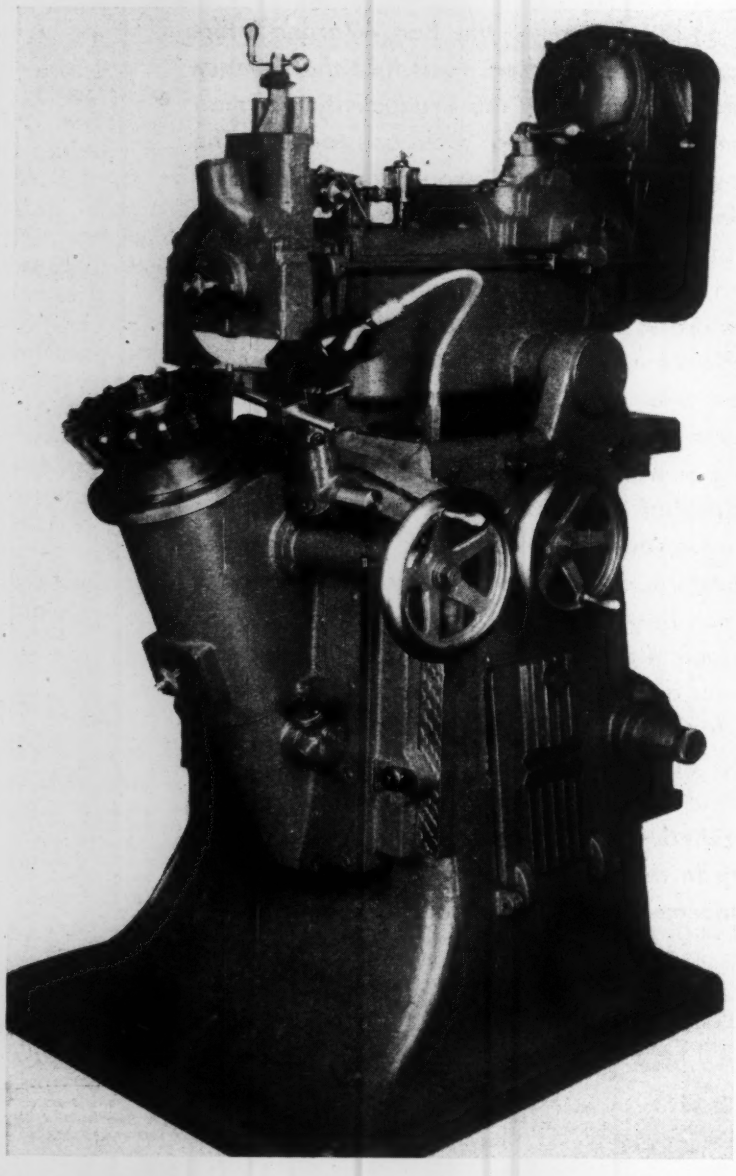
DESIGNS OF THE MONTH

Automatic Cutter Grinder Has Noteworthy Design Features

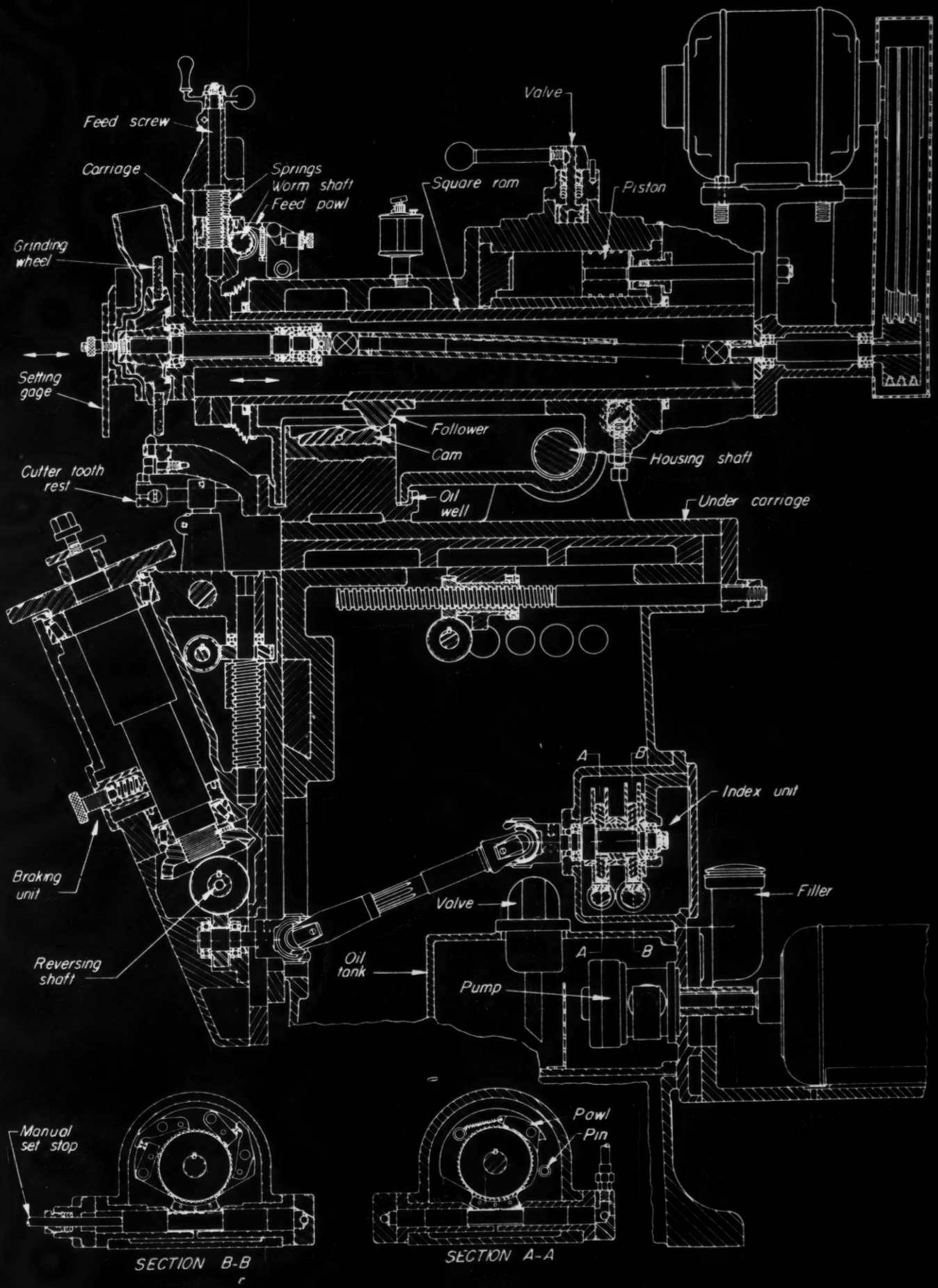
DESIGNED for sharpening face mill cutters, with automatic grinding-wheel dressing for each tooth grind, this machine is powered and cycled hydraulically. It exemplifies exceedingly clever yet practical design and should serve to supply many worthwhile ideas to designers of all kinds of machines.

Referring to the cross section drawing, the grinding wheel and its spindle are mounted on a carriage which provides for vertical adjustment, compensating for wheel wear. When grinding, the grinding wheel travels horizontally, being carried on a square ram which is guided in a housing. A piston and cylinder reciprocate the ram, while the valve serves to start, stop and control the speed of reciprocation.

Bolted to the ram is a follower and, as the ram reciprocates, the follower rides up and down over a cam, causing the wheel to follow the path prescribed by the cam shape (different cams are used for different-shaped cutters). The housing is mounted on a large shaft, carried on roller bearings at each end. All of these units are mounted on a large under carriage which is manually movable horizontally to adjust the machine for different size cutters. A gage is provided for setting the clearance on cutter faces. Sealing of the entire ram and cam mechanism is effected by an oil well which excludes all grit and other foreign matter.



As the grinding wheel travels back off a cutter tooth, it is fed down by a feed pawl which actuates a ratchet wheel mounted on a worm shaft. The ratchet, worm and worm gear (which is also a feed nut) travel in a casting bolted to the front of the ram, and this unit is not adjustable up and down in relation to the ram. When the machine is operating, the feed screw is locked in the carriage end plate and, as the feed pawl turns the ratchet wheel, the grinding wheel is moved toward the diamond, which dresses the wheel before another tooth is ground. Inasmuch as the diamond is stationary and the grinding level of the cutter teeth does not change, each tooth will be the same height and will be ground with a sharp, freshly dressed wheel. The down-feed of the wheel is adjustable down to 0.00005-inch per pass,


Feed screw
Carriage
Springs
Worm shaft
Feed pawl
Valve
Square ram
Piston
Grinding wheel
Setting gage
Follower
Cam
Housing shaft
Cutter tooth rest
Oil well
Under carriage
A
B
Index unit
Braking unit
Valve
Filler
Reversing shaft
Oil tank
A
B
Pump
Manual set stop
Pawl
Pin
SECTION B-B
SECTION A-A

this fineness of feed being made possible by carrying the grinding wheel spindle on two springs which are of sufficient strength to carry the weight of the assembly.

The oil tank, located in the base of the machine, mounts a Vickers pump and combination accumulator-relief valve, set to hold the hydraulic pressure at 100 psi. The index unit is hydraulic and indexes the cutter without the use of index plates. It turns the cutter until a tooth passes the cutter-tooth rest a short distance (about ¼-inch), after which the cutter is turned back and the tooth is held tightly against the rest during grinding. With the grinding wheel running against the cutting edge and the tooth rest against the cutting side of the tooth, the grinding wheel tends to pull the tooth away from the rest, but this is impossible because the hydraulic pressure holds the tooth against the rest regardless of grinding pressure. Functions of the hydraulic indexing unit are indicated in the section drawings *A-A* and *B-B*. The unit shown in section *B-B* indexes the cutter from tooth to tooth, while the unit of section *A-A* returns the cutter and holds it while the tooth is being ground. Pressure is brought into the right-hand ends of the cylinders, forcing the pistons to the left and turning both units clockwise. Unit *A-A* turns until the tail end of its pawl contacts the pin, which stops rotation of the unit and disengages the pawl, leaving the front unit, section *B-B*, free for indexing of the cutter.

Amount of index motion is controlled by a manually set stop (method of setting not shown). At the instant the piston in *B-B* touches the stop, the oil is released from the right-hand end and oil is forced into the left-hand end of both cylinders. Then the pawl of *A-A* engages its ratchet and returns or reverses the cutter, holding a cutter tooth against the rest during grinding. The four pawls in *B-B* are so spaced that only one pawl works at a time. The action obtained is the same as having a ratchet with four times as many teeth working with one pawl, so that the wear is divided among four pawls. By means of a clever design expedient only two springs are used to actuate these four pawls.

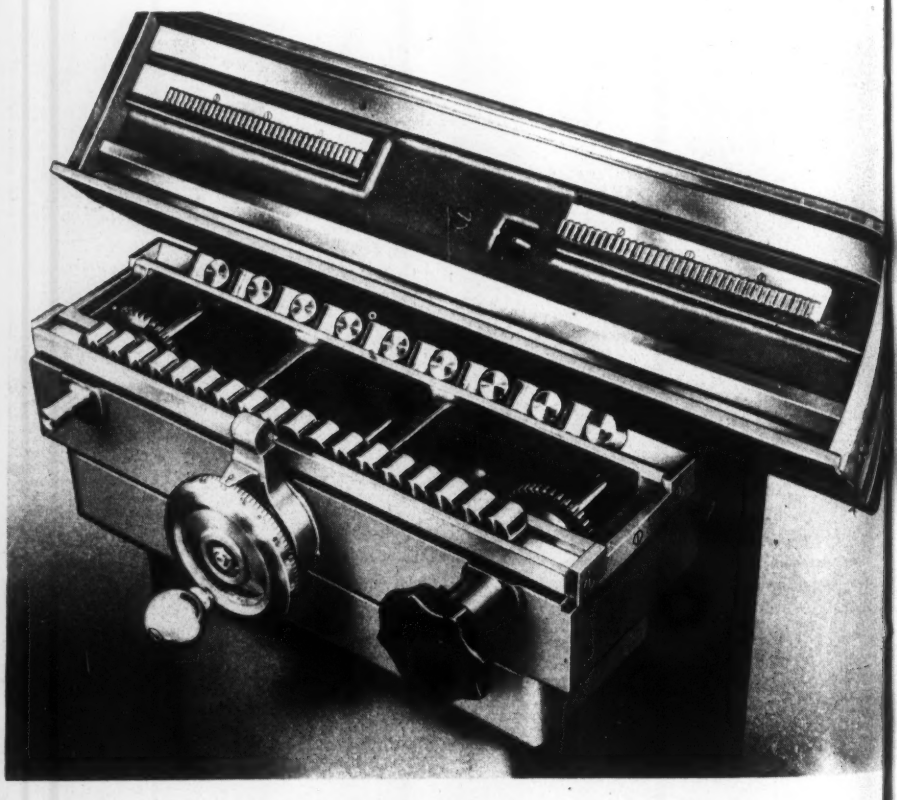
As the pistons in the index unit work, a small amount of oil leaks past them and into the index-unit case, so about one-third of the way up on the side of the case a small tube is attached to return the oil to the tank. In this way, oil level is maintained in the index unit.

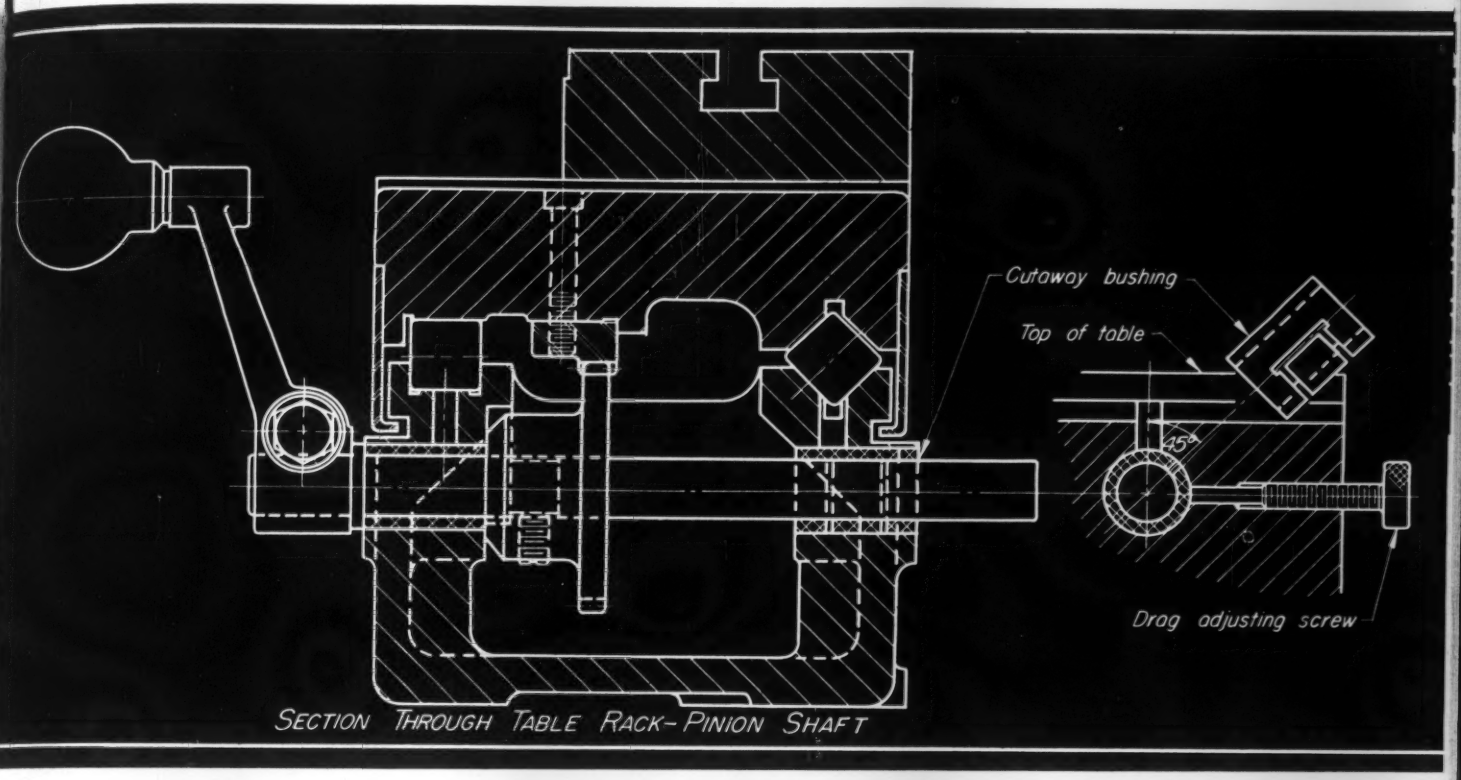
As will be noted, a reversing shaft is provided so that right-hand and left-hand cutters can be ground. On this shaft jaw clutches are located between the helical gear in the center and bevel gears at each end. The shaft is tubular and the clutches are pinned to a small floating shaft inside the tube through elongated slots in the tube. Shifting of the clutches to engage either of the bevel gears is done with a manually-operated eccentric shaft. Manufacturer: Oliver Instrument Co., Adrian, Mich.

Roller Ways in Grinder Cut Table Friction to Minimum

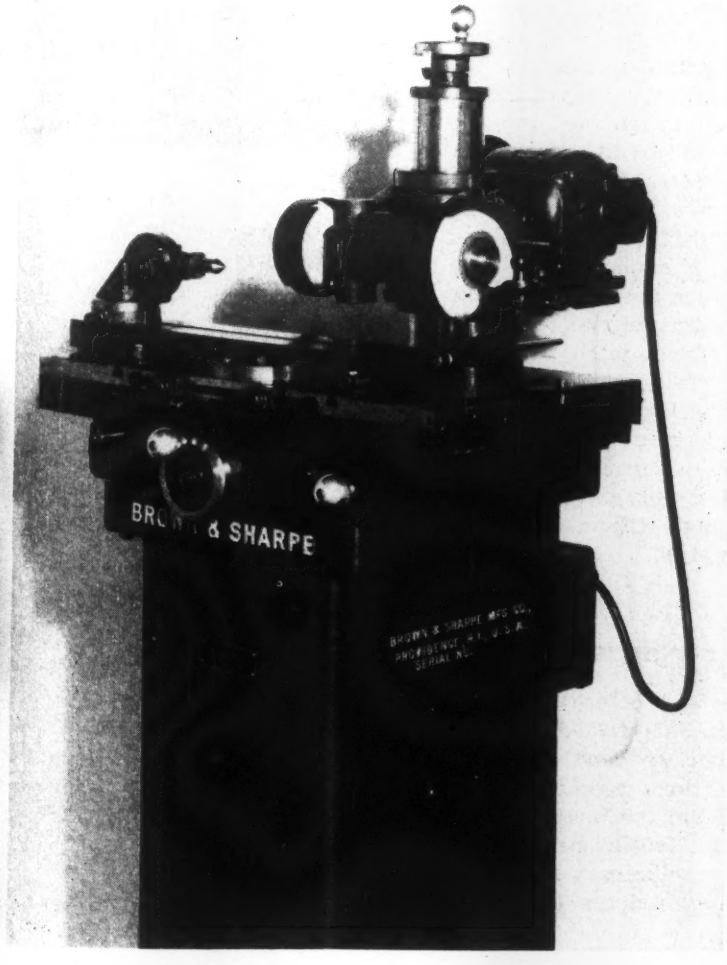
TABLE of this cutter and tool grinding machine slides on 36 precision-ground rollers, the rollers at the front being mounted in a flat channel way, while those in the rear operate in a V-way. Table friction with this design is so slight that an aluminum ball-knob crank has enough unbalance to move the table when the crank is left in a horizontal position.

To take care of cases where a completely free table is not desirable, an adjustable table drag is provided. As the cross section drawing shows, the handwheel or rack-pinion shaft has an inner bushing with a section partially cut away by a U-shaped slot. By turning a knurled adjusting screw, pressure can be exerted on the shaft and any desired drag obtained. There are two table handwheel pinion shafts, one on each side of the machine. Each shaft has an





SECTION THROUGH TABLE RACK-PINION SHAFT



adjustable drag bushing and, to prevent the table from coasting ahead by the amount of the backlash between driving pinion and rack, the drag usually is placed on the hand-wheel shaft which is not being actuated by the operator.

Rollers require no retainer or cage and no return path or circuit is provided. This greatly simplifies the design. In time the rollers will creep and, after many cycles of table movement, they may strike the end plate. Of course when this happens they cannot roll, and further movement of the table in this direction slides the table along the tops of the rollers or slips the rollers on the bed way surface while they turn about their own axes. One such full stroke of the table locates the rollers in their proper position relative to the table and bed, and the table is again ready for another period of free rolling.

Rollers are steel, hardened, ground and lapped, while the V and flat way surfaces are close-grained cast iron with ground finish. Roller diameters are held to a tolerance of 0.0001-inch, and taper in each roller must not exceed 0.00002-inch over the roller length. End faces of the rollers, which prevent skewing, are ground square and lapped. Rollers are lubricated by 300-sec machine oil which covers the lower portions of the bed ways and is carried up and circulated by the rollers as they turn. Manufacturer: Brown & Sharpe Mfg. Co., Providence 1, R. I.

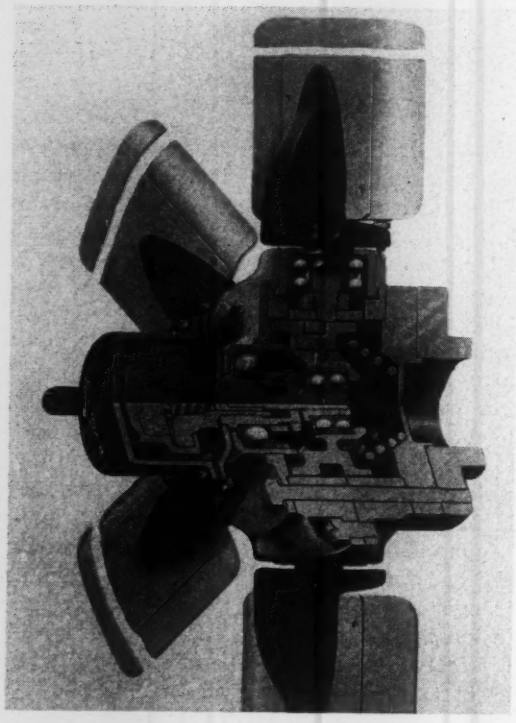
Applications

of engineering parts, materials and processes

Lead-Coated Iron Spout

HOT-DIP lead-alloy coating on engine radiator spout, right, serves to protect the part and also to facilitate joining it to the radiator. Unit is attached to shell by a sweat joint, made possible by the dip, reinforced by rivets. Lead coating, bonded to the cast iron, serves the same purpose as tinning, providing allover corrosion protection.

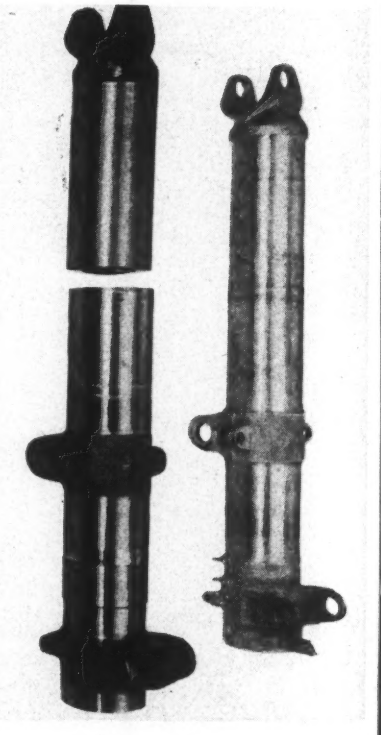
Fan is Thermally Actuated



AUTOMATIC blade-control on the Evans engine-cooling fan, left, is achieved by an integral heat-sensitive unit. As engine load, and therefore engine temperature, changes the Vernatherm thermostatic unit, which is in the coolant circuit, expands or contracts. In so doing, it moves a control rod which is linked to fan blades in such a manner as to turn them on their axis. Engine temperature rise results in high blade pitch; temperature fall results in blade feathering. Temperature of the engine is, as a result, rapidly raised on starting and closely held at point of high efficiency during operation.

Shock Absorbers are Welded Forgings

AIRCRAFT shock-absorber cylinder, shown at right before and after being welded, is made of two forgings machined and butt-welded together. Technique has several advantages: Because of hot working, controlled development of inherent grain growth in the directions needed provides maximum strength and toughness with lighter sectional thickness; reliably predictable heat treatment response assures uniform tensile strengths as high as 200,000 psi with minimum weight; further, preliminary machining of relatively simple individual parts before welding reduces production costs.



Planetary Gear Calculations

. . . simplified by a routine procedure which can be applied to any combination of driving and driven members

By Colin Carmichael
Associate Editor, Machine Design

PLANETARY gear systems offer an almost unlimited variety of possibilities in providing speed ratios between driving and driven members. Only a few of the possible arrangements are well known, although many of the ramifications can provide unique relationships in a simple manner. Among these are the so-called differential drives in which two input shafts drive an output shaft through a planetary unit. For such designs the calculation of speed ratios, while numerically simple, enters the realm where common sense no longer serves as a reliable check. It is therefore essential to have at hand a simple foolproof method which can be applied with confidence. By following the routine set forth in this Data Sheet engineers can quickly determine speed relationships for any conceivable design. The method is not limited to the particular examples shown.

First step is to establish a basic relationship between the speeds of the two sun or ring gears and the planet carrier or arm. Any two of these may be the driving elements, the third being driven. Conventional planetary speed reducers are special cases in which one of the gears is held stationary. In *Fig.* 1 are shown three moving elements, namely, the arm *A*, the smaller sun or ring gear *B*, and the larger sun or ring gear *C*. The fixed frame of reference is denoted by *F*. The intermediate planet pinion or pinions may have any of a wide variety of arrangements. Their arrangement does not affect the basic equation.

The following notation provides the key to the method. Speeds (revolutions per minute) are denoted by *N*, two subscripts being employed to indicate what two elements are involved. For example, N_{af} denotes "the speed of arm *A* relative to the frame *F*." If the subscripts are transposed, N_{fa} denotes "the speed of the frame *F* relative to the arm *A*." It follows that $N_{fa} = -N_{af}$.

Where speeds must be referred to other elements a simple rule is used, which is best illustrated by an example. The speed of *A* relative to *F* may be expressed as follows:

$$N_{af} = N_{ab} + N_{bf}$$

from which is will be noted that (a) the first subscript of the first term, *a*, and the last subscript of the last term, *f*, on the right-hand side correspond to those on the left-hand side (b) the terms on the right are added and (c) the same subscript, *b*, ap-

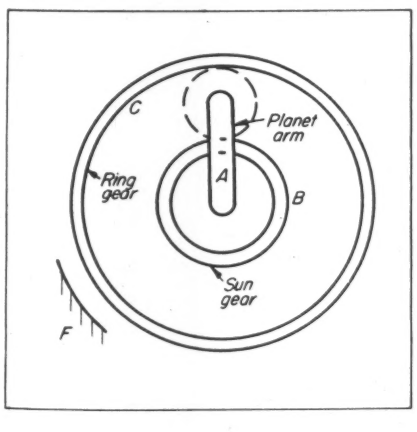


Fig. 1—Left—Basic elements of a planetary drive—planet arm, sun gear and ring gear. Planet pinion, shown dotted, may be replaced by a variety of gear trains

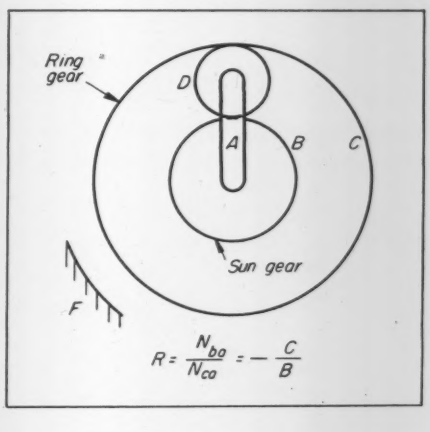


Fig. 2—Right—Simple internal-external planetary. Speed ratio indicated applies when arm A is fixed; planetary ratio is found by substitution in Equation 6

ENGINEERING DATA SHEET

pears immediately before and after the plus sign. The scheme may be extended indefinitely, if necessary, thus:

$$N_{af} = N_{ab} + N_{bc} + N_{cf}$$

Applying the foregoing rules to the general solution of a planetary gear unit, it will be assumed that gears B and C are driving and the planet arm A is driven, *Fig.* 1. It is required to find the speed of the arm A when the other two speeds are known. As a first step the speed of A in terms of the speed of B is written down thus:

$$N_{af} = N_{ab} + N_{bf} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (1)$$

In this equation N_{bf} is known but N_{ab} is not. However, if the equation is rewritten:

$$N_{af} = N_{bf} - N_{ba} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (2)$$

it will be noted that N_{ba} is the speed of gear B relative to the arm A. Since arm A carries the gear centers it will be easy to calculate speed relationships with respect to the arm, and thus correlate the speeds of B and C. Using the symbol R to denote the *speed ratio* between B and C, that is:

$$R = \frac{N_{ba}}{N_{ca}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (3)$$

Equation 2 may be written:

$$N_{af} = N_{bf} - RN_{ca} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (4)$$

In calculating R the proper sign must be used. Thus, if the particular arrangement of planetary pinions gives opposite rotation to B and C, the numerical value of R would be minus.

One more step is necessary to render Equation 4 usable. The speed of C must be related to the fixed frame F, since N_{cf} is one of the known speeds. Applying this rule:

$$N_{ca} = N_{cf} + N_{fa} = N_{cf} - N_{af} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

and combining Equations 4 and 5:

$$N_{af} = N_{bf} - R(N_{cf} - N_{af})$$

Solving for N_{af}:

$$N_{af} = \frac{N_{bf} - RN_{cf}}{1 - R} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (6)$$

Equation 6 is the basic equation which applies to all possible arrangements, the only further calculations required being the evaluation of R in magnitude and sign. This involves simply the computation of a train of gears revolving about fixed centers on the arm A.

Thus, for the example shown in *Fig.* 2, if the arm is held stationary and gear C is rotated, the following holds true:

$$R = \frac{N_{ca}}{N_{ca}} = \frac{N_{ba}}{N_{da}} \times \frac{N_{da}}{N_{ca}}$$

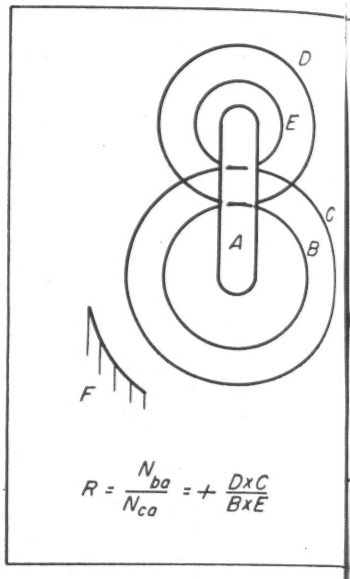


Fig. 4—Compound external planetary. Compound planet pinions permit much higher speed ratios without requiring large differences in gear sizes

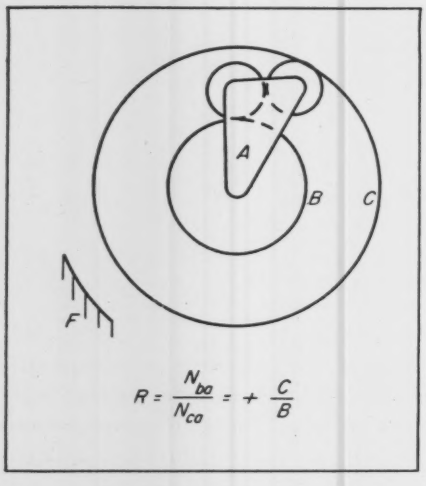


Fig. 3—Left—Simple internal-external planetary with Idler. Extra gear in planet train permits reversing the relative rotation of input and output when gear C is fixed

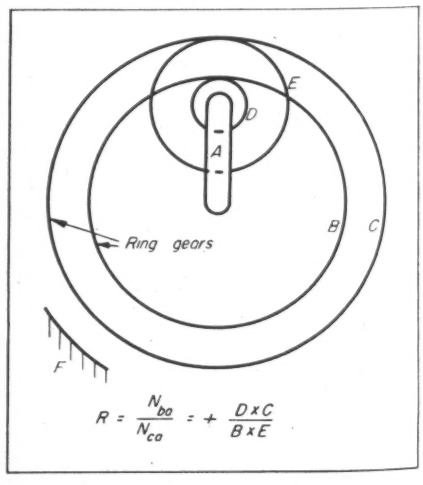


Fig. 5—Right—Compound internal planetary. Like the compound external planetary, this arrangement can provide large speed ratios and reverse output speed if desired

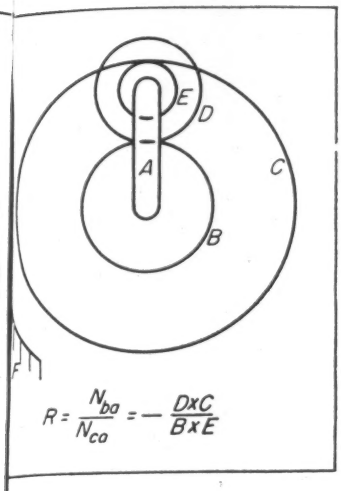


Fig. 6—Compound internal-external planetary. Planet arm A revolves in same direction as the driving gear

Expressing the speed ratios in terms of the numbers of teeth in the gears, using B to denote the number of teeth on gear B, etc.,

$$\frac{N_{ba}}{N_{da}} = -\frac{D}{B} \text{ and } \frac{N_{da}}{N_{ca}} = +\frac{C}{D}$$

Substituting these values in the foregoing equation, $R = -C/B$. For this particular example, then, Equation 6 would become:

$$N_{af} = \frac{N_{bf} + (C/B)N_{cf}}{1 + (C/B)}$$

If either one of the gears is held stationary the corresponding term drops out of the equation, resulting in the well-known relationships for simple planetary speed reducers. Thus, if the ring gear C is held, $N_{cf} = 0$, or if the sun gear B is held, $N_{bf} = 0$.

In *Figs.* 2 to 6 a variety of planetary arrangements is pictured, together with the formulas for R, including the proper sign, in each design. Substitution of the appropriate value of R into Equation 6 gives the complete formula for speed relations. In applying the formulas the correct magnitude *and sign* must be assigned to the input speeds. If one element is held stationary its speed value is zero. If the planet arm A is one of the driving elements, the speed of the driven sun or ring gear can be found by transposing Equation 6.

In addition to the calculation of overall speed ratios it is often necessary to know the pitch-line velocities of gear-tooth engagement. To calculate these velocities the speeds of the gears with respect to the arm must be known. This can be easily found from any one of several of the foregoing equations. Thus, from Equation 2:

$$N_{ba} = N_{bf} - N_{af} \quad \ldots\ldots (7)$$

or from Equation 5:

$$N_{ca} = N_{cf} - N_{af} \quad \ldots\ldots (8)$$

Thus, when the input and output speeds have been determined, the speeds of the gears as they revolve about their centers on the arm, N_{ba} and N_{ca}, can be found.

Compound planetary units permit the addition of extra gears, making possible two or three output speeds from which to select. Examples are illustrated in *Figs.* 7 and 8. In *Fig.* 7 the speed of the extra gear may be found from the relation:

$$N_{gf} = N_{ga} + N_{af} \quad \ldots\ldots (9)$$

If R_1 denotes the speed ratio between G and B, that is:

$$R_1 = \frac{N_{ga}}{N_{ba}} \quad \ldots\ldots (10)$$

then:

$$N_{gf} = R_1 N_{ba} + N_{af}$$

Substituting for N_{ba} from Equation 7:

$$N_{gf} = R_1 (N_{bf} - N_{af}) + N_{af}$$
$$= R_1 N_{bf} + (1 - R_1) N_{af} \quad \ldots\ldots (11)$$

Thus, when N_{bf} and N_{af} are known, the additional output speed N_{gf} is readily calculated. A similar re-

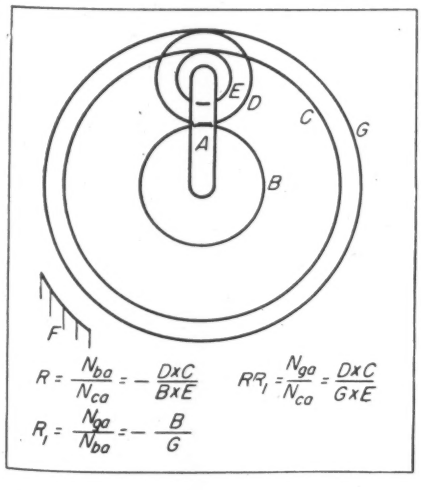


Fig. 7—Left—Compound internal-external planetary with two ring gears. Extra gear permits choice of output speeds, either from planet arm or from one of the gears

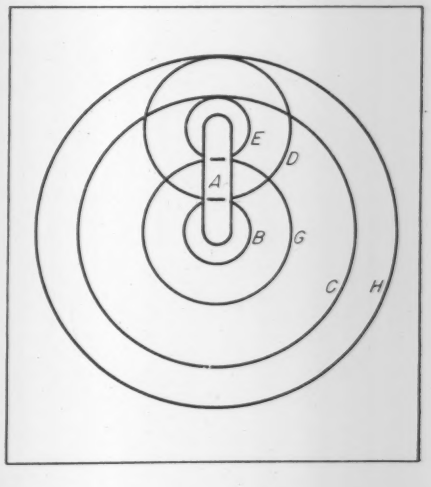


Fig. 8—Right—Compound internal-external planetary with two sun and two ring gears. This is the general case of which the designs in the previous figures are special

lation may be established for the second additional gear, H, in *Fig.* 8.

EXAMPLE 1: For the design shown in *Fig.* 9 the number of teeth in the gears and pinions are as follows: $B = 23$, $C = 28$, $D = 28$, $E = 18$, and $G = 79$. With the sun gear C rotating at 1000 rpm, it is required to find the speeds of the planet arm A (the output) and the ring gear G with sun gear B locked to the frame.

Inasmuch as $N_{bf} = 0$, Equation 6 becomes

$$N_{af} = -\frac{RN_{cf}}{1-R} = \frac{RN_{cf}}{R-1}$$

where $R = N_{ba}/N_{ca}$. With the planet arm fixed, $N_{ba}/N_{ca} = -C/B = -28/23 = -1.217$, therefore $N_{af} = -1.217 \times 1000/(-1.217-1) = 550$ rpm. To find the speed of ring gear G, Equation 11 becomes:

$$N_{gf} = (1-R_1)N_{af}$$

where $R_1 = N_{ga}/N_{ba} = -B/G = -23/79 = -0.291$, therefore $N_{gf} + [1-(-0.291)] \times 550 = 710$ rpm.

Now if the sun gear B is released and the the ring gear G is locked to the frame, what will be the speeds of the planet arm A and the sun gear B? These could be found by substituting in Equations 6 and 11 and solving simultaneously for N_{af} and N_{bf}. However, a more direct method is to go back to the derivation of Equation 6 and rewrite it with N_{gf} replacing N_{bf} and the ratio $R_2 = N_{ga}/N_{ca}$ replacing R. It will be noted that the only change is to replace b with g in the subscripts. Then

$$N_{af} = \frac{N_{gf} - R_2N_{cf}}{1-R_2} = \frac{-R_2N_{cf}}{1-R_2}$$

since $N_{gf} = 0$. Then $R_2 = +C/G = 28/79 = 0.354$, and $N_{af} = -0.354 \times 1000/(1-0.354) = -550$ rpm. The speed of sun gear B can now be solved from either Equation 6 or 11. Using the latter equation:

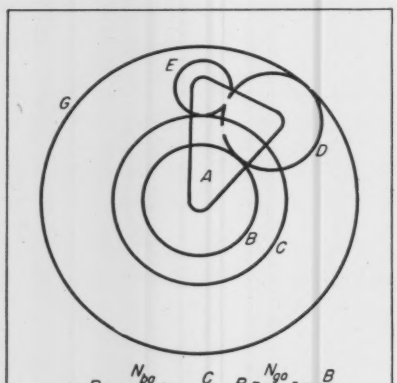


Fig. 9 — Compound internal - external planetary with idler planet pinion. Used in the new Buick transmission, this is one example of the sort of variations that can be made in the basic designs illustrated in the foregoing

$$N_{bf} = \frac{-(1-R_1) \times N_{af}}{R_1} = \frac{-(1.0291) \times (-550)}{-0.291} = -1940 \text{ rpm}$$

Finally, when both the sun gear B and the ring gear G are released the transmission is in neutral and the planet arm A receives no driving torque. Under this condition, with the arm at rest and gear C rotating at 1000 rpm, what will be the speeds of B and G? Inasmuch as the unit no longer operates as a planetary, Equations 3 and 10 give the answer. Thus, $N_{ba} = RN_{ca} = -1.217 \times 1000 = -1218$ rpm and $N_{ga} = R_1 \times N_{ba} = (-0.291) \times (-1218) = 354$ rpm.

EXAMPLE 2: In an aircraft propeller pitch-changing drive (MACHINE DESIGN, April, 1945, Page 135) the arrangement is similar to that shown in *Fig.* 7, with the ring gear G fixed to the housing and sun gear B driving, the output being taken from ring gear C. The numbers of teeth in the gears are as follows: $B = 12$, $C = 109$, $D = 54$, $E = 43$, $G = 120$. It is required to find the speed reduction ratio between B and C.

This may be calculated by solving simultaneously Equations 6 and 11, noting that $N_{gf} = 0$ for this design. However, a more direct method is to apply the rule for referring speeds to members that are known or can be calculated. Thus, in order to calculate speed ratios it is necessary to refer to the planet arm, A, that carries the gear centers, and to the fixed gear G. The desired speed reduction ratio may then be written:

$$\frac{N_{bf}}{N_{cf}} = \frac{N_{ba} + N_{ag} + N_{gf}}{N_{ca} + N_{ag} + N_{gf}} = \frac{N_{ba} - N_{ga}}{N_{ca} - N_{ga}} = \frac{N_{ba}/N_{ga} - 1}{N_{ca}/N_{ga} - 1}$$

From *Fig.* 7 the speed ratio $N_{ba}/N_{ga} = -G/B = -120/12 = -10$ and:

$$N_{ca}/N_{ga} = \frac{G \times E}{D \times C} = \frac{120 \times 43}{54 \times 109} = 0.877$$

hence:

$$\frac{N_{bf}}{N_{cf}} = \frac{-11}{-0.123} = 89.5$$

Torque relationships may be readily established for any planetary arrangement once the speed ratios have been determined. The calculations are based on two principles—total external torque acting on the unit must be zero, and input and output powers must be equal, neglecting losses. Torque summation is:

$$T_a + T_b + T_c + \text{etc.} = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots (12)$$

where the list includes all driving, driven, and fixed elements, and the proper signs are applied.

For each member the power transmitted is proportional to TN, hence the power equation can be written:

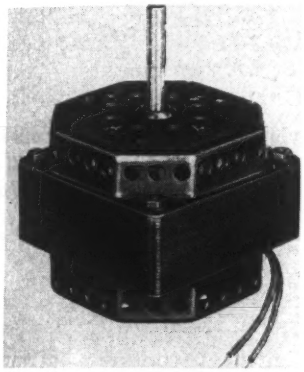
$$T_aN_{af} + T_bN_{bf} + T_cN_{cf} + \text{etc.} = 0 \quad \ldots\ldots\ldots\ldots\ldots (13)$$

In applying Equations 12 and 13 scrupulous attention must be paid to the sign of the numerical values. If this is done the system cannot fail to yield the correct answer; if it is not done, torque calculations can be completely off.

new parts and materials

To obtain additional information on these new developments see Page 263

Shaded-Pole Induction Motor



Four-pole shaded-pole induction motor rated 12 oz-in. has been especially designed for quiet and smooth operation. This new RM-4 motor is made for 60-cycle, 115-volt operation either clockwise or counterclockwise and operates at a free speed of 1720 rpm. Features of the unit include special locating and locking means for covers in order to assure accuracy of rotor alignment. This, combined with dynamic balancing of rotors results in freedom from vibration and noise. Motor has self-aligning end-thrust and sleeve bearings, wick lubrication and forced-air cooling. Unit measures 3⅜ inch square and 2⅞ inch high; shaft is 5/16-inch in diameter and weight is 4½ lb. Manufacturer: General Industries Co., Elyria, Ohio.

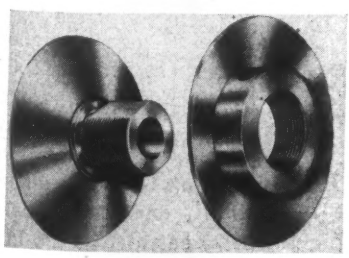
For further information circle MD 1 on card Page 263

New Thermoplastics

Two new resins in the Geon line of vinyl chloride-vinylidene chloride copolymers offer increased thermoplasticity and lowered softening point. The plastics, which are identified as Geon resins 204 and 205, have tensile strengths of 6875 and 5885 psi, respectively, and are recommended for such applications as piping and battery separators. Manufacturer: B. F. Goodrich Chemical Co., Rose Bldg., Cleveland 15.

For further information circle MD 2 on card Page 263

Variable-Pitch Sheave



Variable-pitch sheave, for use with fractional horsepower drives, provides for fine adjustment by the use of 20 threads per inch instead of the usual 16. Of steel construction throughout, this model FHP sheave is made in eight sizes to cover the pitch diameter range of 1.9 to 4.7 inches. Each has a one-inch adjustment. Three standard bores are available: ½, ⅝ and ¾-inch. The smallest of these is provided with set screw and the larger two sizes have both keyway and setscrew. Manufacturer: Dodge Mfg. Co., Mishawaka, Indiana.

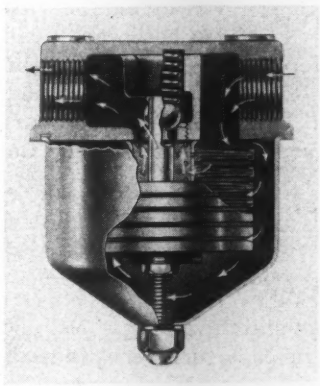
For further information circle MD 3 on card Page 263

Electronic Speed Control

Variable-speed device known as the Flexoid electronic speed control is designed for use with electric motors up to one hp in size. Unit is recommended for use where especially smooth regulation is desired since it provides stepless regulation from zero to full motor speed. Manually controlled by means of a dial, it operates from 220-volt, single-phase current and measures 13½ inches high, 13 inches wide and 6½ inches deep. Manufacturer: The Smith Power Transmission Co., 1545 E. 23d St., Cleveland 14.

For further information circle MD 4 on card Page 263

All-Metal Oil Filter



Full-flow type filter is designed for use with lubricating, hydraulic or fuel oils. Filtering is achieved by passage of the fluid through metallic screening having openings only 0.0032-inch in width. As a result of the small opening size and the "filter aid" effect of accumulated dirt, particles smaller than 5 microns are said to be collected. Features are low pressure drop, approximately 2 psi, and ease of cleaning. Models available range in capacity from 1.7 to 45 gpm, with fittings threaded ⅛ to 1½ inch FPT. Manufacturer: Air-Maze Corp., 5200 Harvard Ave., Cleveland 5.

For further information circle MD 5 on card Page 263

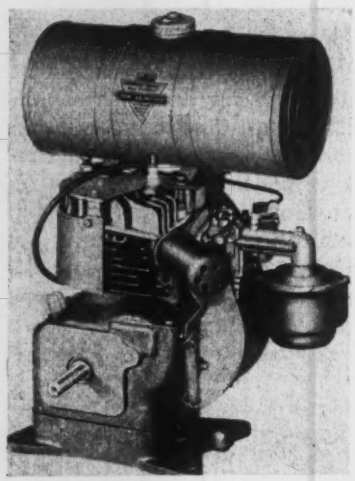
Electronic Control Relays

Line of all-purpose electronic control relays is available in 2, 3 or 4-pole, normally-open or double-throw contact types. They have standard silver contacts with a capacity of 15 amperes at 24 volts dc or 11 volts ac, noninductive. The type PO has a coil rating of up to 10.5 amperes at 25 or 60 cycles, 220

volts; or 2.5 watts at 120 volts dc. Type POY, intended for operation direct from plate of a vacuum tube or other low-power circuit, is a semisensitive, dual-coil relay for dc use only. Its coil rating is one watt at 110 volts dc. Both relays are available in hermetically sealed models. Four-pole units measure 2¼ by 1⅞ by 2 3/16 inches; three pole units measure 1⅝ in the last dimension. Manufacturer: Allied Control Co. Inc., 2 East End Ave., New York 21.

For further information circle MD 6 on card Page 263

Four-Cycle Air-Cooled Engine



Gasoline engine rated 2½ to 3 hp at 2600 to 3600 rpm is available in gear reductions of 6:1, 4:1 or 2:1. The unit, known as model 1000, is an air-cooled, L-head engine having a single cylinder. Features of the new prime mover include splash-type lubrication, aircraft - type magneto, and cast-iron cylinders. Various bases are available for mounting on different types of equipment; oil filler is located on top of crankcase and drainage may be had at either end. Unit is suitable for portable equipment. Manufacturer: Clinton Machine Co., Clinton, Mich.

For further information circle MD 7 on card Page 263

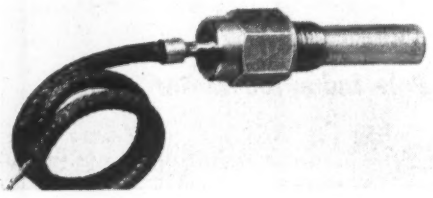
Acid Resisting Stainless Steel

Sulphuric-acid resisting stainless steel No. 20, previously available only in cast form, may now be had in wrought shapes. Comparable in analysis to Durimet 20, the new formulation possesses all of the advantages of the cast material and, in addition, that of workability. The steel is available in hot rolled and drawn rounds, rectangles, squares and hexagons; in centerless-ground rods, in tubing and in hot and cold strip up to 8 inches in width and 0.140-inch in thickness. Alloy is also resistant to nitric and dilute hydrofluoric acid as well as all vegetable juices. Manufacturer: The Carpenter Steel Co., Reading, Pa.

For further information circle MD 8 on card Page 263

Thermal Limit Switch

Thermal control and limit switch has a control range from −60 F to + 600 F, and an ac or dc current capacity of 2 amperes at 28 volts. The unit is designed to warn of excessive temperatures in fluids used, for example, in cylinder blocks, exhaust manifolds, etc. Switch is made with ⅜-24, class 2 thread,

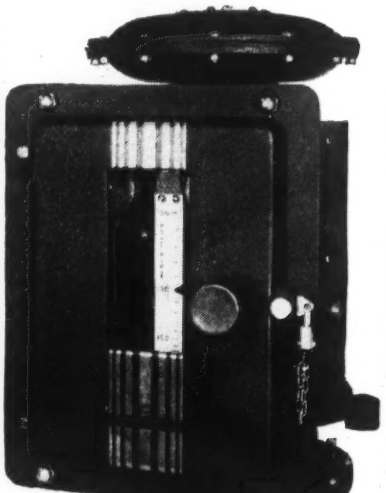


is readily installed. Materials used are cold-rolled steel, brass or stainless steel, depending on application. Manufacturer: Control Products, Inc., 306 Sussex St., Harrison, N. J.

For further information circle MD 9 on card Page 263

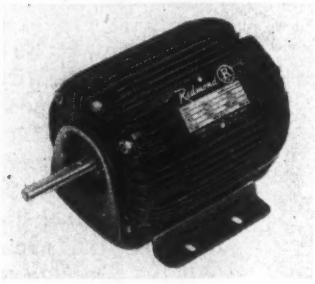
Air-Fuel Ratio Controller

Combustion control unit for use with industrial furnaces is a balance-type instrument easily adjusted for operation with fuels of various Btu content. The instrument employs electric - motor valve drives, and maintains air-fuel ratio constant at the desired control point. It can also provide automatic variation of ratio, increasing or decreasing the per cent of air at reduced fuel flows. Manufacturer: Leeds & Northrup Co., 4934 Stenton Ave., Philadelphia 44.



For further information circle MD 10 on card Page 263

Fractional-Horsepower Motors



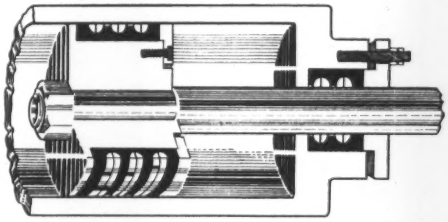
Six-pole shaded-pole motors designated Type Y are designed for operation at 1000 rpm under full rated load. Line includes ventilated models in sizes up to 1/15-hp and totally enclosed models in sizes up to 1/20-hp. All of the units have die-cast cases with finned cooling surfaces, welded rotors, filled rotor slots, and large porous bearings. Oil reservoirs have capacity of 6 cubic inches. Motors

may be operated in any position without change in construction; they are particularly recommended for driving fans. Manufacturer: Redmond Co., Inc., Owosso, Mich.

For further information circle MD 11 on card Page 263

Metallic Packing

New floating-seal packing of the metallic type is suitable for sliding as well as rotating shafts. It will hold unusually high pressures, one segment, for example, withstanding 1500 psi. Each unit of the packing consists of several metal rings molded within a ring of synthetic material. Packing is recommended



for use with such machines as compressors, steam units, hydraulic units, oil or acid pumps, brine pumps and refrigeration units. Manufacturer: Rodpak Mfg. Co., 1315 Natoma St., San Francisco 3.

For further information circle MD 12 on card Page 263

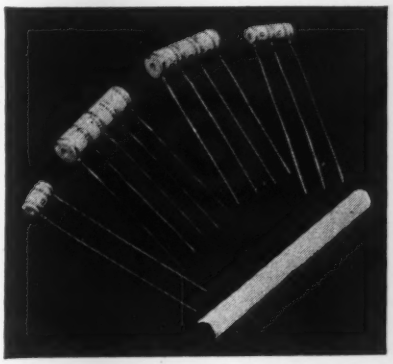
Variable Autotransformers

Line of high-capacity Variacs includes two new models, one rated at 115 volts 20 amperes and the other 230 volts, 2.3 kva. The 115-volt, 3.45 kva, model is identified as type V-20M, while the 230-volt style is known as type V-20HM. Output voltage of the transformers is continuously adjustable from zero to 17 per cent above line voltage. Both models are supplied with case and terminal-box cover. Overall dimensions are 7⅞ by 9⅝ by 5½ inches, weight is 22¾ lb. Manufacturer: General Radio Co., 275 Massachusetts Ave., Cambridge 29, Mass.

For further information circle MD 13 on card Page 263

Precision Resistors

Designed for miniature equipment where space and precision are at a premium, new resistors are available in four styles. Type 136, smallest yet produced by this manufacturer, has a maximum power rating of 0.25 watt and measures 1 5/32 inches in length and ⅜-inch in diameter. Maximum resistance is 150,000 ohms. Types 137, 133 and 134 are 2, 3 and 4-section units with 2, 3 and 4 leads, respectively, and with 0.25 watts maximum load per section. Maximum resistance of 550,000 ohms is available in type 133. The



resistors have 2-inch tinned-copper leads and can be suspended by the wires for mounting. Manufacturer: Shallcross Mfg. Co., Jackson and Pusey Aves., Collindale, Pa.

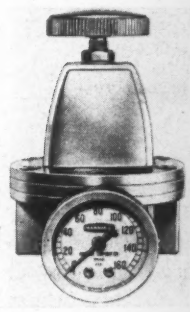
For further information circle MD 14 on card Page 263

Conveyor Belt for Hot Material

Conveyor belt using glass fabric instead of cotton has been developed for carrying relatively high-temperature materials. Examples include hot lime at 450 F, and hot foundry molding sand. Advantage of the use of glass lies in its resistance to loss of strength at the temperatures encountered, cotton losing strength rapidly under severe operating conditions. Two weights are recommended: Four-ply for light service and five-ply for wider, longer belts. Covers recommended are either 5/32-inch top and 1/32-inch back, or 3/16-inch top and 1/32-inch back. Manufacturer: B. F. Goodrich Co., Akron, Ohio.

For further information circle MD 15 on card Page 263

Air Regulating Valve



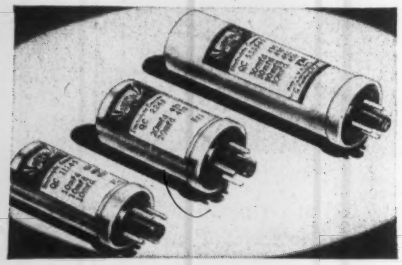
Regulator for use with primary air pressures up to 150 psi can be set to maintain reduced pressures ranging from 5 to 125 psi. Feature of this valve is the free floating stem. It is possible to reduce the delivered pressure under dead-end conditions by turning the adjustment screw. This is desirable when the applied pressure must be reduced without wasting gas. Other features of the regulator include the incorporation of a fine-mesh bronze strainer, the use of Nylon valve seats and the all-over use of noncorrosive materials. Unit,

known as the Air Warden, is made in two models. Type RD-1 is designed for use with ⅜-inch piping and type RD-2 is suitable for use with ½-inch connections. Manufacturer: Hannifin Corp., 1101 S. Kilbourn Ave., Chicago 24.

For further information circle MD 16 on card Page 263

Plug-In Capacitors

Line of plug-in capacitors is suited for experimental applications where quick changes are required. Designated as type QC, the new units are hermetically sealed in round aluminum containers and are provided with a four-pin octal base mounting which permits ready removal or replacement in standard tube sockets. The units are available in a popular range of capacity and voltage combinations. Manufacturer: Cornell-Dubilier Electric Corp., South Plainfield, N. J.



For further information circle MD 17 on card Page 263

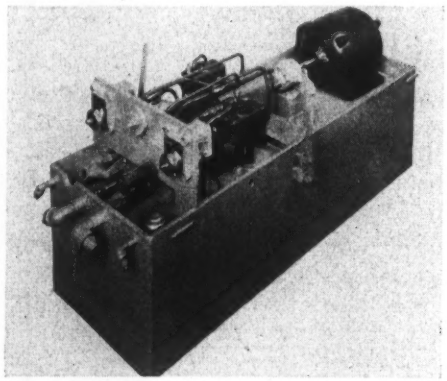
Speed-Change Control

Variation in speed of V-belt drives up to a maximum ratio of 10:1 is provided by the Toledo variable-speed transmission. Single-arm type will provide speed control up to 3:1, while the double-arm type will provide speed ratios to 10:1. Models are available to accommodate both A and B-section belts and will deliver power within the limits of belt strength. Manufacturer: Toledo Timer Co., 2224 Albion St.,Toledo, Ohio.

For further information circle MD 18 on card Page 263

Hydraulic Power Unit

Completely self-contained power units for actuating hydraulically-driven machinery, Hufford power units are available in sizes from one to fifty horsepower. Each unit consists of an assembly of motor, pump, valves and flow controls mounted above a hydraulic reservoir and filter system. A cover completely encloses the unit, preventing entrance of dust and other foreign material. Built-in selector valves enable both forward and reverse motion of the machine to which the power is applied, and the flow controls permit speed variation of the driven mechanism over a wide range. Units are suitable for both manual and automatic operations, and can be furnished in a variety of types from a simple motor-driven pump to complicated systems for all-automatic operation. Manufacturer: Hufford Machine Works, Redondo Beach, Calif.



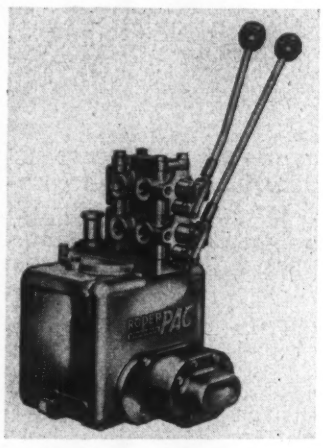
For further information circle MD 19 on card Page 263

Adhesive for Silicone Rubber

New bonding agent known as G-E adhesive No. 12509 is designed for use in fastening silicone rubber to itself or glass, metals and ceramics. The material possesses properties similar to silicone rubber, remaining flexible and resilient over temperatures ranging from −70 F to +520 F. Adhesive will withstand continuous temperature of 350 F and has successfully been used for securing silicone gaskets direct to oven doors. Manufacturer: General Electric Co., chemical department, Pittsfield, Mass.

For further information circle MD 20 on card Page 263

Hydraulic Power Units



Series 1400 hydraulic power units are compact devices recommended for powering hydraulic cylinders or motors operating at about 800 psi. Two models are now available: Type 3 is designed for belt or chain operation and is to be driven at 500-1000 rpm. At 560 rpm it is rated 4½ gpm and 800 psi. Type 4, for direct-drive operation, is intended to be driven at 250-500 rpm; at 340 rpm its rating is also 4½ gpm, 800 psi. Both models feature the use of either one

or two independently operable spool-type control valves. Advantage of these valves includes automatic return to "hold" position when handle is released. Manufacturer: Geo. D. Roper Corp., Rockford, Ill.

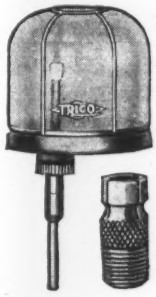
For further information circle MD 21 on card Page 263

Variable-Speed Motor

Designed to develop constant, maximum torque at all speeds, new electric motor is variable in speed from 1500 to 6000 rpm. Lever at side of unit controls a pair of centrifugal vibrating contacts which adjust the speed and also serve to start and stop the motor. Completely self contained, the motor measures 5 by 7 by 9 inches, is rated 1/5-hp operating on 115-volt a-c power. Universal wound, unit is readily reversible. Manufacturer: Guernet Electrical Machinery Inc., Box 196, Meriden, Conn.

For further information circle MD 22 on card Page 263

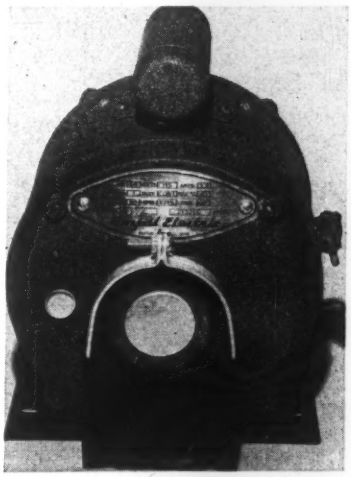
Unbreakable Bottle Oilers



Visible, unbreakable bottle oiler designed for limited space has offset feed spout making it possible to mount unit where clearance is as little as ¾-inch. Oiler automatically lubricates wick or waste-packed bearings. For direct feed a rod slides on the rotating shaft and slight vibration or movement releases oil from the plastic bottle. Made in 2 or 4-oz capacities, the unit is installed using ⅛-inch pipe-thread adapter. Manufacturer: Trico Fuse Mfg. Co., 2948 N. 5th St., Milwaukee 12.

For further information circle MD 23 on card Page 263

Capacitor-Type Motors

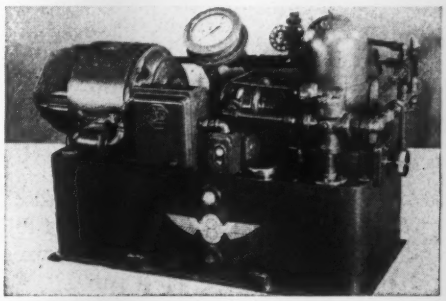


Fractional-horsepower capacitor-start induction motor is mounted on vibration damping trunnion base. Among features of motor are extremely close tolerances used in manufacture and careful dynamic balance. Smooth and troublefree operation is said to be thus assured. Bearings are diamond-bored bronze, winding is heavy varnish coated, and trunnions are live rubber to reduce noise transmission. Manufacturer: Royal Electric Inc., Jamestown, Ohio.

For further information circle MD 24 on card Page 263

Hydraulic Power Units

Self-contained power unit providing hydraulic power for rolling mills, eliminates need for intricate manifolding and accessories. The model PP-113 power pack is designed to provide and maintain a steady uniform pressure on calender rolls regardless of varying thick-



ness and seams in the material. Unit is self-contained, consisting of reservoir, motor and pump, magnetic starter, valves, gage and accumulator. Manufacturer: Greer Hydraulics Inc., 454 18th St., Brooklyn, 15, N.Y.

For further information circle MD 25 on card Page 263

Self-Aligning Ball Bearings

Double-row bearing designed for light loads and medium speed is so constructed that it compensates for misalignment and deflection on shafts. This sealed

unit has grooved outer race and extended inner race for locking on shaft. Having machined rather than ground races, it is relatively inexpensive. Manufacturer: The Heim Co., Fairfield, Conn.

For further information circle MD 26 on card Page 263

Totally-Enclosed Motors

Line of totally enclosed, fan-cooled, wound-rotor motors have tube-type air-to-air heat exchangers. High efficiency of these exchangers permits reduction in size of motor and, because of simplicity of design, reduced cleaning problem. Units can be built with stainless-steel ventilating tubes, fans and end plates and cast-iron terminal boxes and collector-ring enclos-

ures. The improved motors are suitable for all fields of industry since they can be adapted readily to suit special requirements. Manufacturer: Allis-Chalmers Mfg. Co., Milwaukee.

For further information circle MD 27 on card Page 263

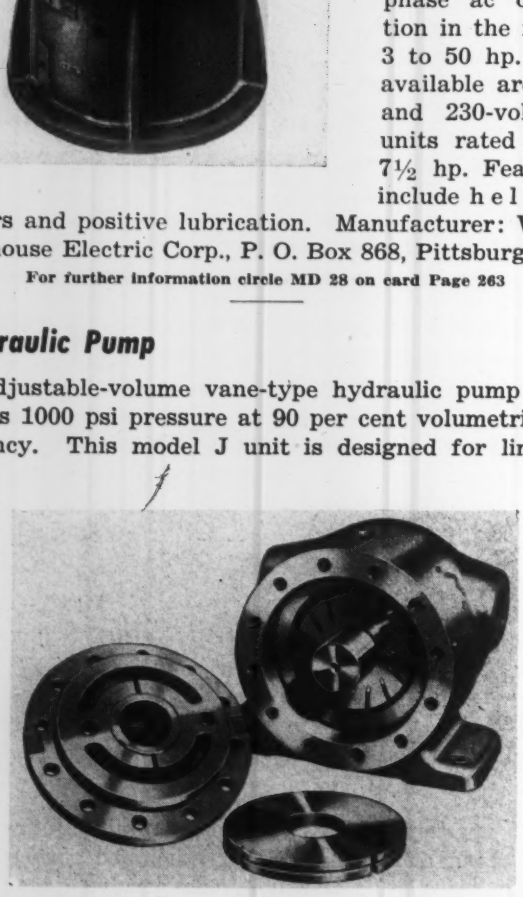
Vertical-Shaft Gearmotors

Line of vertical gearmotors is made in nine reduction combinations ranging from 7.61:1 to 38.9:1. Motors available include 220, 440 or 550-volt sizes for 3-phase ac operation in the range 3 to 50 hp. Also available are 115 and 230-volt dc units rated 3 to 7½ hp. Features include helical gears and positive lubrication. Manufacturer: Westinghouse Electric Corp., P. O. Box 868, Pittsburgh 30.

For further information circle MD 28 on card Page 263

Hydraulic Pump

Adjustable-volume vane-type hydraulic pump produces 1000 psi pressure at 90 per cent volumetric efficiency. This model J unit is designed for limited

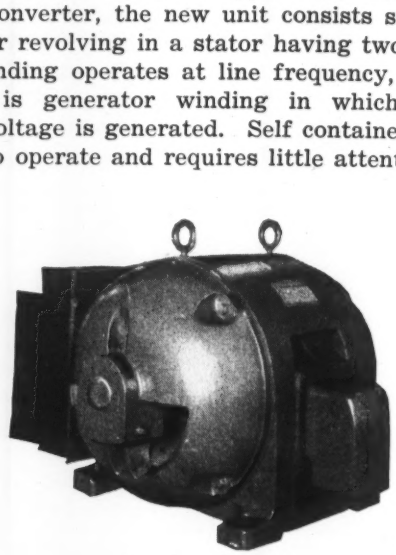


space applications, seating area being only 6 by 6 inches. Pump is available with either foot or flange mounting, and features wear-compensated valves and self lubrication. Manufacturer: Racine Tool and Machine Co., Hydraulic Div., Racine, Wis.

For further information circle MD 29 on card Page 263

Power Supply for High-Speed Motors

Self-excited frequency converter for powering high-speed a-c motors provides 3-phase power at frequencies from 360 to 2160 cycles, delivering ½ to 150 kw. Called the Tri-Clad type AKC inductor frequency converter, the new unit consists simply of a cast rotor revolving in a stator having two windings. Input winding operates at line frequency, secondary winding is generator winding in which high-frequency voltage is generated. Self contained, the unit is easy to operate and requires little attention. Con-



trol usually consists only of a magnetic switch with conventional thermal-overload relay on the input side. Manufacturer: General Electric Co., Schenectady 5, N. Y.

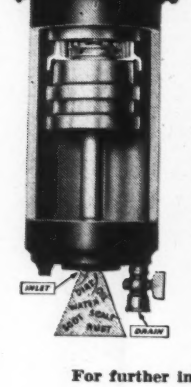
For further information circle MD 30 on card Page 263

Roller-Bearing Pillow Block

New pillow block consists of flange-type self-contained roller bearing attached to sturdy base by a metal strap. Unit provides efficient, low-cost means of supporting shafts. Pillow blocks may be mounted at any angle, are kept dust free by the flanged construction of the outer casing. Sizes now available accommodate shafts up to to one inch in diameter. Units may be mounted upright or at any angle. Manufacturer: The Heim Co., Fairfield, Conn.

For further information circle MD 31 on card Page 263

Air Separator



Separator for removing moisture and impurities from steam, gas or air has no screens, filters or moving parts. The model E separator is mechanical in operation, requiring no attention other than draining condensate as it accumulates. Noncorrosive metals are used throughout. Unit measures 8 inches in diameter and 15 inches high, has standard 1½-inch pipe connections and is rated at 250 cfm. No adjusting or regulating of the unit is said to be necessary. Manufacturer: Chicago Mfg. and Distributing Co., 1928 W. 46th St., Chicago 9.

For further information circle MD 32 on card Page 263

High-Speed Plug-In Relay

Plug-in relay having an operating time of one millisecond or less is mechanical rather than electronic in action. This type 162 relay is available in one size only — 110-volts, ½-ampere dc—and is a single-pole, single-throw device housed in a standard metal radio-tube container. Two coils are incorporated in the design, the first being an operating coil functioning on 3 volts, and the second a hold-in coil operating on 6 volts. Addition of the hold-in coil makes the Millisec relay useful for photoflash and other applications where ultra-high speed closing time followed by a long holding-in period is required. Mfr: Stevens-Arnold Co., 22 Elkins St., South Boston, Mass.

For further information circle MD 33 on card Page 263

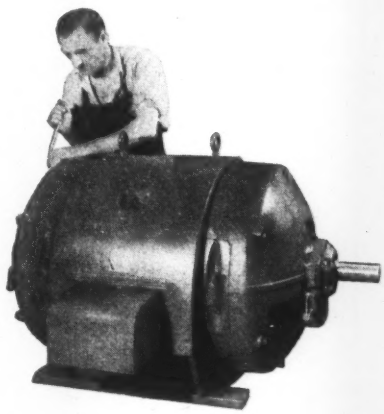
High-Pressure Hydraulic Pump

Twin-piston type high-pressure hydraulic pump has pressure range from 100 to 4000 psi. Having power requirements (at 4000 psi) of 5 hp, the unit has a displacement of 0.389 cu in. per revolution and operates at 600-900 rpm. At the latter speed, capacity is 1.375 gpm. The pump measures 4 inches in diameter and 5¾ inches in height and weighs 18 lb. Manufacturer: Blackhawk Mfg. Co., Milwaukee 1.

For further information circle MD 34 on card Page 263

Squirrel-Cage Induction Motor

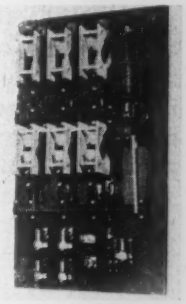
Two-pole, squirrel-cage induction motor is intended for applications such as driving high-speed pumps, fans and compressors. Rated 200 to 700-hp at 3600



rpm, NEMA Class B starting (normal torque, low current), the motor is of drip-proof construction and has a fabricated-steel frame. Among the features are: Centrifugally-cast sleeve bearings, split bearing brackets and phos-copper brazed cage winding. Manufacturer: Electric Machinery Mfg. Co., 821 Second Ave., S. E., Minneapolis 13, Minn.

For further information circle MD 35 on card Page 263

Automatic Transfer Switches



Automatic transfer switch for use in machine controls or circuit-breaker switchboards has normal and emergency power-source poles mounted one above the other in order to decrease width of installation. Electromagnetic, the switch is made in ratings from 75 to 1000 amperes for ac or dc application. For built-in applications, units are furnished with front-connected services at top and bottom and with back-connected terminals for load circuits at center. They are also available with all terminals back connected. Manufacturer: Automatic Switch Co., 391 Lakeside Ave., Orange, N. J.

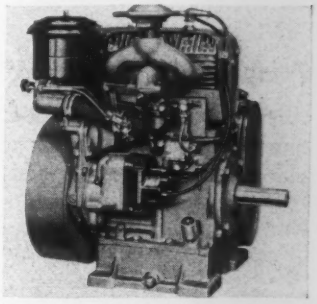
For further information circle MD 36 on card Page 263

Variable-Speed Control

Electronic control for use with variable-speed transmissions provides linear speed control despite characteristics of the speed changer. This packaged-unit type control will maintain speed setting despite variations in load or characteristics of transmission, and will control speed ratios between several different systems where desired. Unit, known as the Dyna-Link, consists of a master control and a speed-measuring generator. When control is set, pilot motor of unit is energized, adjusting speed change until actual output speed corresponds to speed setting. If speed variations occur, controller automatically detects error and corrects it. Manufacturer: Yardeney Laboratories Inc., 105-107 Chambers St., New York 7.

For further information circle MD 37 on card Page 263

Two-Cylinder Air-Cooled Engine

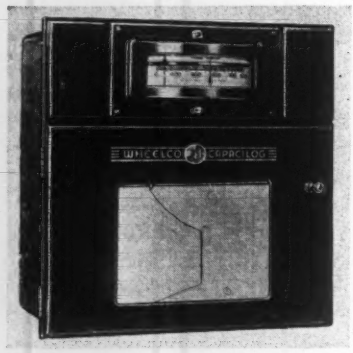


Two-cylinder, four-stroke cycle, air-cooled engines are available in two models covering ratings of 8.3 to 13.3 hp. Model TE, operating at 1600 to 2600 rpm, is rated 8.3 to 11.2 hp, while model TF operating at the same speeds is rated 9.9 to 13.3 hp. Both engines weigh 220 pounds. The units can be supplied with electric starter and generator, clutch take-off assembly and clutch reduction unit. Standard design requires gasoline

for fuel. However, types may be furnished to burn other fuels. Manufacturer: Wisconsin Motor Corp., 53d and Burnham Sts., Milwaukee 14.

For further information circle MD 38 on card Page 263

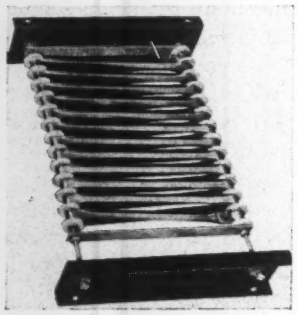
Strip-Chart Recorder



New recorder, known as the Capacilog, is designed for the measurement indication, control and permanent record of variables in processing systems. Operating on 115 or 230-volt, 25, 50 or 60-cycle power, the Capacilog is available in a wide number of models utilizing a variety of control systems. Maximum power consumption is 60 watts. Manufacturer: Wheelco Instruments Co., 847 W. Harrison St., Chicago.

For further information circle MD 39 on card Page 263

Electrical Resistors



Metallic resistors are recommended for applications involving continuous loads, vibration and shock. The elements, known as Loopohm, consist of channeled resistance ribbon, of uniform section, formed in a series of loops. Standard resistors are available with continuous current capacities from 20 to 110 amperes and a wide range of resistance values. Manufacturer: Ward Leonard Electric Co., Mount Vernon, N. Y.

For further information circle MD 40 on card Page 263

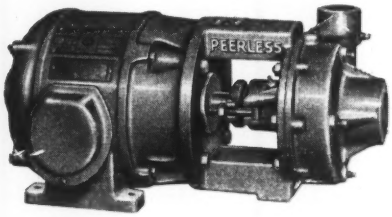
Temperature Controller

Stepless input controller and temperature indicator has been developed for use with furnaces and other electrical heating devices. Incorporated in the unit is a sensitive thermostatic switch controlled by a knob on the instrument panel. Knob can be set to regulate the current input into the heating equipment within the range 5 to 100 per cent time on. Instrument contains a sensitive pyrometer and is automatically compensated for changes in line voltage, maintaining constant watt-hour input and even temperature regardless of voltage fluctuations. Manufacturer: Thermo Electric Mfg. Co., 480 W. Locust St., Dubuque, Iowa.

For further information circle MD 41 on card Page 263

General-Duty Centrifugal Pumps

Single-stage, end-suction centrifugal pumps are available in two types. Type PE is an integrally coupled unit in which pump and driving motor are



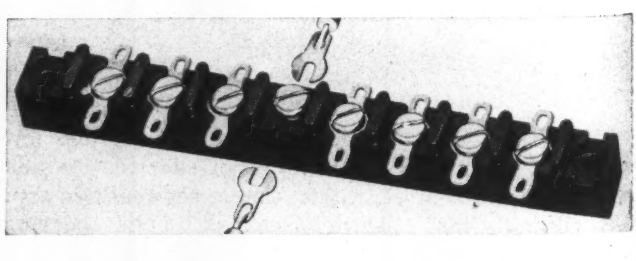
mounted on the same motor footing. Type PB is a belt-driven unit with the pump frame and bearing mounts integrally cast with footing. Units, known as Fluidyne, are suitable for moving fluids in capacities up to 1000 gpm against heads up to 270 ft and may be mounted in any position. Manufacturer: Peerless Pump Div., Food Machinery Corp., 301 W. Ave. 26, Los Angeles 31.

For further information circle MD 42 on card Page 263

Plastic Terminal Block

Featuring a terminal lock-in arrangement, new block is said to eliminate the danger of spade lugs pulling out. Solder-type lugs recessed into the body of the block, are rigidly held in place against a molded-in shoulder by the tightening of the binder screws. Since base of the phenolic block is solid, screws are entirely insulated from ground. The terminal unit is particularly designed for conditions of space limitation; a ten terminal block, for example, measures only 5⅞-inches long, 13/16-inch wide and ⅝-inch high. Models are available with up to 18 terminals. Manufacturer: Curtis Development & Mfg. Co., 1 N. Crawford Ave., Chicago 24.

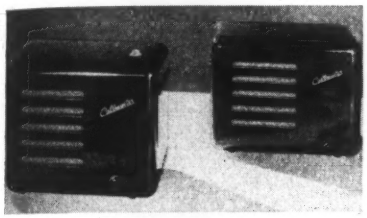
For further information circle MD 43 on card Page 263



engineering dept equipment

In order to obtain additional information on this new equipment see Page 263

Intercommunication Set



Electronic intercommunication set providing two-way voice transmission between two stations consists of a "master" and a "sub." The packaged unit is so designed as to be easily installed by the user. Known as the '47 Callmaster, the instrument features plastic cabinets and high sensitivity and power output. Manufacturer: Lyman Electronics Corp., 12 Cass St., Springfield, Mass.

For further information circle MD 44 on card Page 263

Electric Tachometer

Hand-type tachometer weighing three pounds is designed to give direct readings of linear speeds from 10 to 10,000 fpm, and rotational speeds from 100 to 100,000 rpm. The instrument consists of two elements

—the head which is placed in contact with the moving object, and the indicating unit to which the head is attached by flexible wires. Inasmuch as there is no gear transmission used in changing speed ranges, shifting may be accomplished while the spindle is rotating; vibrations from the rotating device being tested do not affect the reading of the instrument. The tachometer can be used to measure rotation both clockwise and counterclockwise. It is furnished in a carrying case. Manufacturer: General Electric Co., special products division, Schenectady, N. J.

For further information circle MD 45 on card Page 263

Drawing Pencil

Graphite drawing pencils known as "Castell" not available since shortly after the beginning of the war, are again available. The pencils, made with graphite milled in Bavaria, will be made in 18 grades of lead all encased in American cedar. Manufacturer: A. W. Faber-Castell Pencil Co. Inc., Dickerson & Bittman Sts., Newark, N. J.

For further information circle MD 46 on card Page 263

Dry Cleaner for Drawings

Preparation known as Draw-Kleen is used to clean drawings and to keep them from being smudged in preparation. Sprinkled on the drawing while work is being done, the material absorbs perspiration, surface dirt and pencil smudges. It is brushed off easily when the drawing is completed. Used to clean completed drawings it is sprinkled on and rubbed over the surface with the palm of the hand. Manufacturer: The Craftint Mfg. Co., 1615 Collamer St., Cleveland.

For further information circle MD 47 on card Page 263

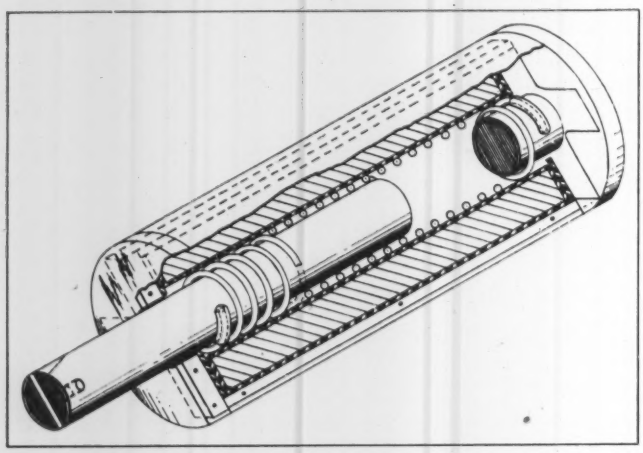
Refillable Drafting Pencil

New refillable drafting pencil has nonslip rubber clutch which will not nick or score leads. Leads are therefore not weakened and breakage is said to be greatly reduced. Long barrel permits use of any full-length drafting lead; knurled grip prevents slipping in fingers. A feature of the pencil is the adjustable cap which indicates the hardness of lead being used in the pencil. Hardness identification is easy and improper selection is thereby avoided. Manufacturer: Charles Bruning Co. Inc., 4754 Montrose Ave., Chicago 41.

For further information circle MD 48 on card Page 263

Noteworthy Patents

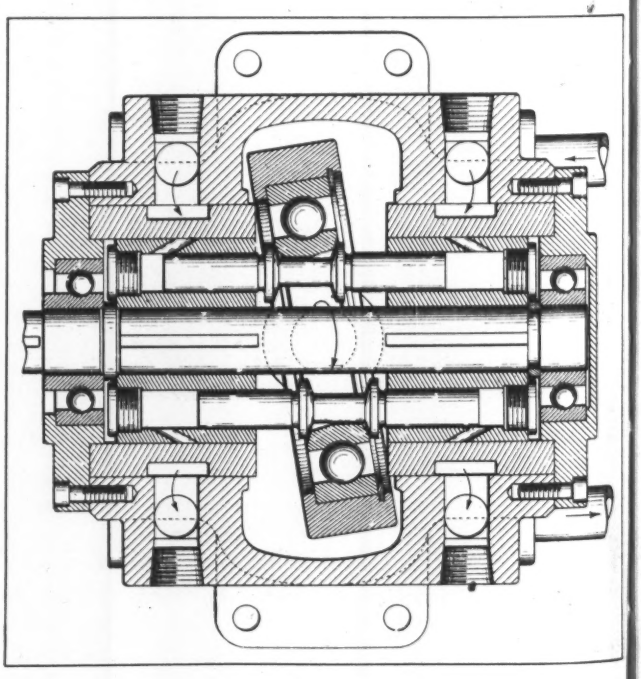
UNIFORM PULL along the entire length of a solenoid plunger is achieved with a new design covered in patent 2,426,745 while chattering, magnetic hum, wattage-loss, and overall weight are considerably reduced. Supporting construction for the winding and plunger guide structure are such that there is no flux shielding or heat build-up within the solenoid. Assigned to Vickers, Inc., by Anthorny J. Porskievies, the solenoid employs iron laminations to provide the outer circuit, a laminated pole piece, and a laminated plunger construction.



AUTOMATIC SUCTION selector for mobile hydraulic systems insures full supply of oil to the suction inlet of the pump regardless of the tilt of the oil reservoir. Covered in patent 2,427,859 this selector utilizes multidirectional feeder pipes on the tank bottom which connect to a like number of radial ports in a vertical tubular valve. A weight secured to the upper part of the valve stem causes the valve stem to register with the suction feeder pipe which taps the lowest part of the oil sump at all times. Patent assigned to the Jeffrey Mfg. Co. by R. K. Jeffrey.

STEP-BY-STEP electric motor for applications such as in tachometer, counter and similar drives, utilizes a combined electromagnetic and mechanical action to obtain regular, stepwise rotation. Operated by separate electrical impulses of direct current the motor provides a fast and positive indexing motion in clockwise or counterclockwise direction. Utilizing magnetic fields to draw trailing armature poles into alignment, the mechanism thereby also depresses detent balls against their springs by cam action and upon interruption of the current spring energy advances the cam through the remainder of the step. Capable of exerting considerable power and speed, the unit can be designed to suit requirements by varying the number of field coils used. Covered by patent 2,428,882 assigned to The Electric Tachometer Corp., by Petrus van Santen Kolff.

LARGE CAPACITY hydraulic pump of comparatively small physical size is detailed in a patent by Adam Ziska under number 2,430,753. Arranged so that operation transmits no pressure or thrust to the main bearings, dual pistons of the pump are located in a cylindrical bank about the main drive shaft. Actuation of the piston stroke is accomplished by means of a special ball-bearing yoke which, when pivoted from a neutral position, creates a swash-plate action to oscillate the pistons. All parts are simple, symmetrical units which can be produced readily on standard manufacturing equipment.



POWDER METALLURGY

Ledaloyl SELF LUBRICATING BRONZE

Low Cost, Self-Oiling, Long Life

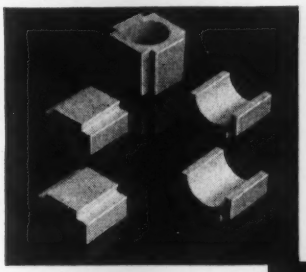
SLEEVE BEARINGS

MANUFACTURERS of many types of equipment can gain these advantages by using Johnson LEDALOYL Bronze for bearings and small parts. LEDALOYL is a product of powder metallurgy, manufactured by an exclusive process using pre-cast, pre-alloyed bearing bronze.

LEDALOYL bearings and parts are molded, under pressure, to required shape and size, thus eliminating expensive machining operations. When quantities of any size are used, the unit cost is surprisingly low.

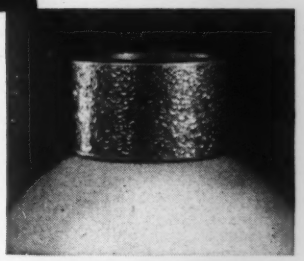
Another valuable feature is the self-lubricating action of LEDALOYL. Myriads of tiny, evenly spaced pores serve as miniature oil wells. When the bearing is in use the oil is metered to the shaft . . . when at rest the oil is absorbed by these pores. This action provides adequate lubrication at all times . . . preventing wear and in most cases eliminating the expense and bulk of other lubricating aids. Service records show long, trouble-free operation on many types of installations.

When you are looking for ways to reduce cost and to increase efficiency in your bearing applications . . . call on Johnson Bronze. We manufacture all types of Sleeve Bearings . . . we are ready to serve you now.



Low Cost

Odd shaped parts and self-aligning bearings when produced by conventional methods require considerable machine work, as compared with one forming operation in LEDALOYL.



Self-Oiling

Place a LEDALOYL bearing on a lighted lamp bulb. Watch the oil sweat from the pores. Remove the bearing and the oil is reabsorbed. This illustrates the self-lubricating and oil retaining action.

Long Life

On a recent "break-down" test, LEDALOYL Bearings operated week after week on a schedule of nine hours a day for five days a week. The accumulated hours of test run represented a normal use of about five years, yet there was no sign of wear in the bearings or on the shaft.

JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS

BRANCHES IN 20 INDUSTRIAL CENTERS

JOHNSON BRONZE COMPANY • 525 South Mill Street . . . New Castle, Pa.

Torque Converter Drive

(*Concluded from Page 124*)

ring gear to the housing. In all cases the planet carrier is the driven member, except in neutral when the clutch and both brake bands are released and no driving effect results.

A hydraulic system is provided for the purpose of circulating fluid through the converter and oil cooler and for furnishing power to operate the clutch and brake bands of the planetary transmission. Pressure is maintained in the system by either or both of two gear-type pumps, the front one driven by the engine and the rear one by the propeller shaft. The front pump functions at starting, low speed, and in reverse; the rear pump is needed if for any reason the engine has to be started by pushing or towing. At speeds above 45 mph the rear pump takes over the whole job, the discharge from the front pump being simply recirculated. The complete transmission requires 11 quarts of a special oil, of which 8 quarts are contained in the converter while operating, the remainder being required for lubrication and for the hydraulic controls. The entire supply is continuously circulated through the oil cooler.

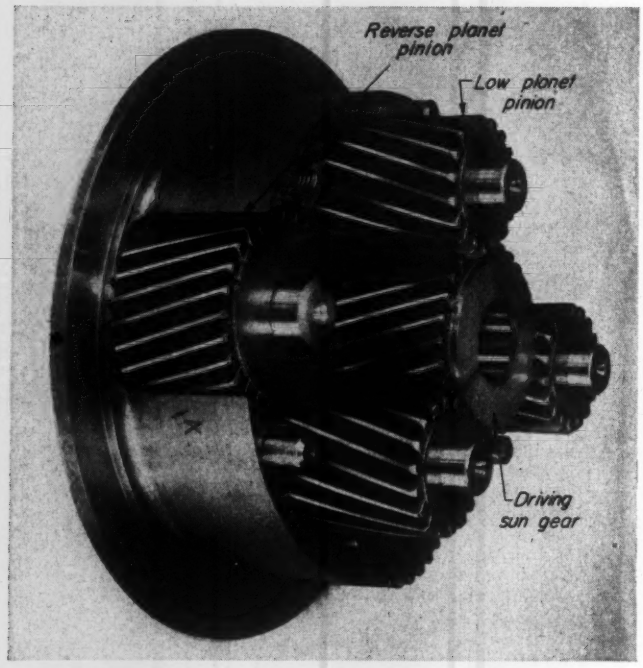


Fig. 6 — Above — Principal elements of planetary unit, with reverse gear and portion of planet carrier removed

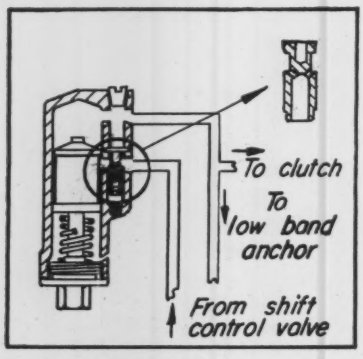


Fig. 7—Left—High accumulator permits clutch to fill rapidly but meters the discharge during final engagement

Key components of the hydraulic circuit are: A master pressure regulator valve which controls the oil pressure, keeps the converter full of oil under pressure while operating, and prevents oil from draining out of the converter when idle; and a shift control valve, operated by the driver from the shift quadrant on the steering column, which directs fluid pressure to the appropriate clutch or brake cylinder.

Supplementing this basic system are several additional components. One of these, the "high accumulator", *Fig.* 7, has the special function of insuring smooth clutch engagement when going into direct drive. Oil flows to the accumulator at 90-psi pressure from the shift control valve, enters around the neck of the pump valve spool and is permitted to flow rapidly and in large volume through the accumulator ports to the back of the clutch piston. A small port in the head and two ports through the shoulder of the valve spool allow oil to bleed above and below the spool so as to equalize pressure and prevent spool movement while the clutch cylinder is being filled.

How Smooth Engagement Is Achieved

Rapid filling continues up to the first stage of engagement. However, as soon as the clutch is filled and actual engagement begins, back pressure is built up in the accumulator, forcing the accumulator piston against a spring. In its downward movement the accumulator piston closes the large upper port, forcing the oil still in motion to meter through the small port in the head of the pump valve to complete the final engagement of the clutch. This metering provides very smooth final engagement.

When shifting out of direct drive, oil pressure from the shift control valve is released and the line from the accumulator to the control valve is opened to the sump. At this point the pressure of the heavy clutch release spring comes into play, forcing the oil in the clutch to return to the sump. Instant release of the clutch is assured by the proper proportioning of the orifice and bleed holes in the dump valve, previously noted. As the oil is dumped, pressure in the clutch line falls off rapidly, permitting the accumulator spring to return to its normal position.

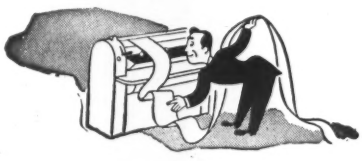
When shifting into emergency low, which requires the application of the low brake band, a similar unit called the "low accumulator" is brought into action. In addition, a pressure booster is employed to raise the pressure to 180 psi for operating the low band.

Another special provision is means for completely draining the "clutch-apply" chamber, the situation in reverse and neutral being such that the low range drum and direct drive clutch piston revolve at much higher speeds than the input shaft. The resulting centrifugal force in any oil remaining in the chamber could partially engage the clutch. A ball check and vent are fitted in the piston head to drain oil from the chamber, with special provisions for preventing, detrimental suction.

NEW! THE OZALID STREAMLINER

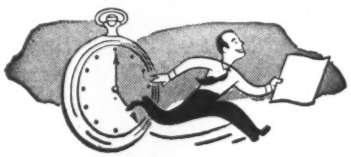
- Reproduces your engineering and architectural drawings in seconds—also your typed, printed, photographic material.
- Moderately priced . . . designed for the thousands of drafting rooms that want these 5 EXTRA VALUES in Printmaking at no extra cost—

1. EFFICIENCY! You always get positive (not negative) prints direct from your tracings . . . prints that are sharper, brighter, much easier for you to read, check, and make notations on.



You produce these without waste of material or waste of motion. Your tracings can be up to 42 inches wide, any length . . . and can be printed either on rolls of Ozalid sensitized paper or on cut sheets of matching size.

Your prints are always delivered dry, ready for immediate use . . . after just two simple operations—Exposure and Dry Development.



2. SPEED! ONLY 25 seconds to reproduce your standard-size tracings, specification and data sheets, etc.

3. ECONOMY! An 8½ x 11-inch reproduction costs you *one cent;* 11 x 17 inches, two cents . . . and so on. The Ozalid Streamliner soon pays for itself . . . in time, labor, and dollars saved.

With it, you can also effect amazing short cuts in design. For example—eliminate redrafting when changing obsolete drawings . . . combine the details of separate tracings on *one* print . . . re-

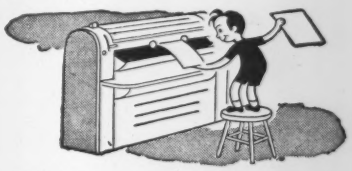
claim old or worn tracings . . . make transparent overlays in different colors.

4. VERSATILITY! You can reproduce the lines and images of any original in black, blue, red, sepia, or yellow . . . on paper, cloth, foil, film, or plastic.

Simply use the Ozalid sensitized material you think best for job at hand; e.g., use identifying colors for prints of separate departments or operations . . . DRYPHOTO to produce beautiful con-

tinuous-tone prints from film positives (which can be made from any negative) . . . OZAPLASTIC to produce oilproof, waterproof prints for shop or field use. *All* prints are made in same fast, economical manner.

5. SIMPLICITY! NOW—printmaking is an easy desk job, automatic in practically every detail.



Anyone can feed originals and sensitized material into the Ozalid Streamliner. Prints are delivered on top, stacked in order—within easy reach of the operator, who does not have to leave her chair.

You can install your Streamliner anywhere; it requires only 11 square feet of floor space.

Write today for free, illustrated booklet . . . showing all the ways you can use the new OZALID STREAMLINER . . . and containing actual reproductions — like those you can make.

Gentlemen: DEPT. 33
Please send New Ozalid Streamliner booklet . . . containing reproductions of drawn, typed, printed, and photographic material. No obligation.

Name_____

Position_____

Company_____

Address_____

OZALID

DIVISION OF
GENERAL ANILINE AND FILM CORPORATION
Johnson City, New York

Ozalid in Canada
Hughes Owens Co., Ltd., Montreal

Assets to a Bookcase

Fundamentals of Engineering Mechanics

By Alvin Sloan, associate professor of applied mechanics, Massachusetts Institute of Technology; published by Prentice-Hall Inc., New York; 379 pages, 6 by 9 inches, clothbound; available through MACHINE DESIGN, *$5.65 postpaid.*

Noteworthy addition to the literature on mechanics, this volume covers the elements of statics and dynamics in a concise, yet painstaking manner. The designer wishing to acquaint himself with this subject for the first time could have few better texts recommended. Discussions are extremely careful. Verbal ambiguities, so common to most engineering writing are virtually eliminated. The volume is, as indicated by the title, not addressed to the advanced student of mechanics. However, many engineers will undoubtedly find the general coverage useful. Of especial interest will be the discussions of the principle of virtual work and Mohr's circle as applied to moments and products of inertia.

□ □ □

Fundamentals of Plastics

By H. M. Richardson, consulting engineer, DeBell and Richardson, and J. Watson Wilson, former director of instruction, New Haven YMCA Junior College; published by McGraw-Hill Book Co., New York; 483 pages, 5½ by 8¾ inches, clothbound; available through MACHINE DESIGN *$5.00 postpaid.*

This book is an outgrowth of courses presented at the New Haven Junior College during the war under the auspices of the Engineering, Science and Management War Training Program. Consequently, it represents the work and knowledge of not two men but of thirteen and the research and authority of such organizations as General Electric, Bakelite Corp. and American Cyanamid.

The volume may be considered to consist of four parts. The first deals with the fundamental relationships of chemical structure to physical properties. Seven tables showing chemical structure, physical and chemical characteristics, and applications of basic plastics should be of great help to engineers in evaluating the properties that may be expected of plastics and in determining the type of plastic to use for specific conditions.

Second section discusses the six classes of plastics, devoting a chapter to each major group. These groups are: Condensation resins, cellulosics, vinyls, elastomers, polymers and silicones. The third section covers commercial manufacturing methods and the last section discusses evaluation of properties and quality.

It will be noted that the book is not basically one concerned with design details. Rather, it is intended to offer a good understanding of the plastics materials now available in order that the engineer may, knowing their advantages and limitations, use them intelligently.

□ □ □

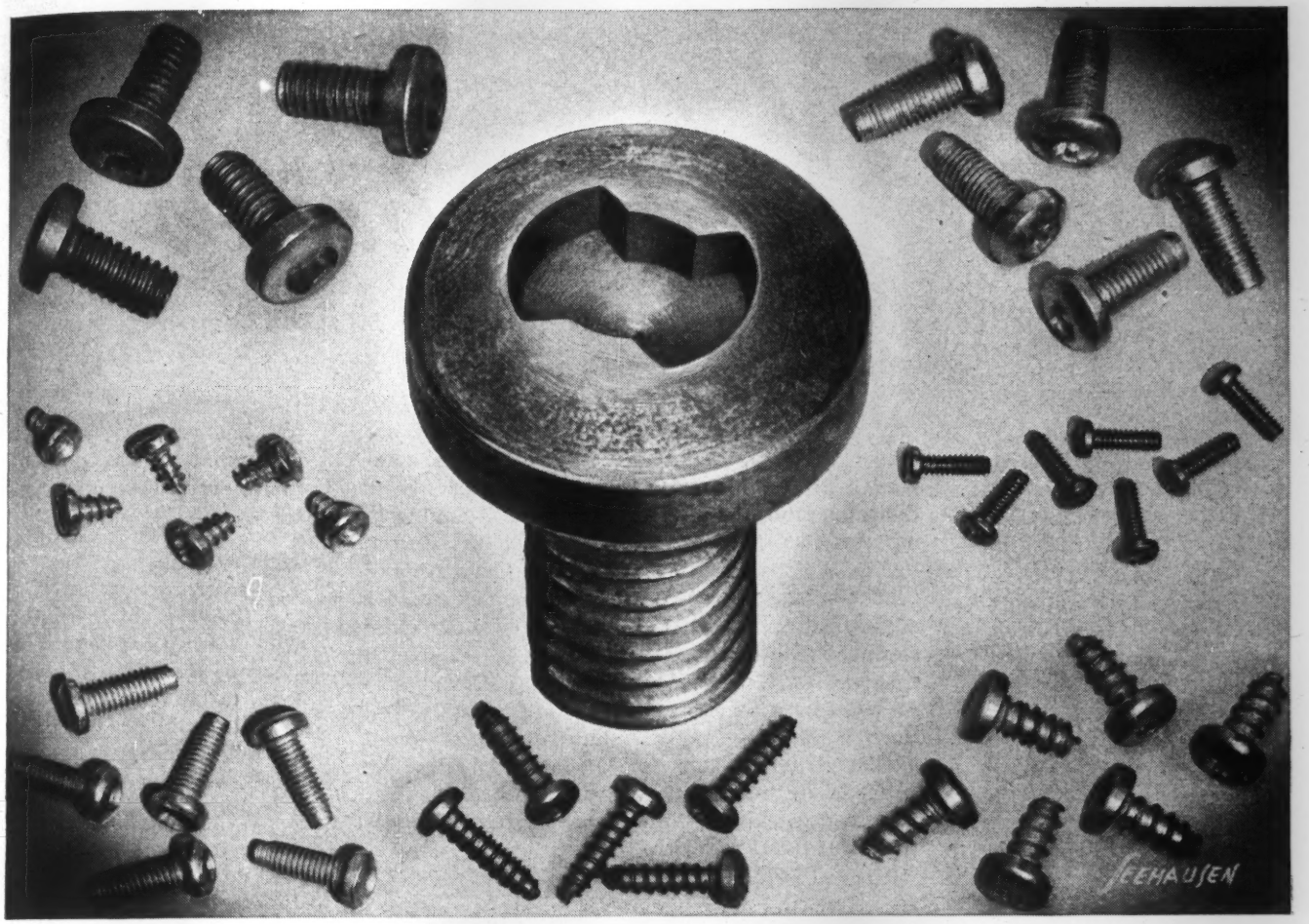
Electronics for Industry

By W. I. Bendz, Westinghouse Electric Corp.; published by John Wiley & Sons Inc., New York; 501 pages, 5½ by 8¼ inches, clothbound; available through MACHINE DESIGN, *$5.00 postpaid.*

This book offers a careful, nonmathematical discussion of the fundamentals of electronics of value to engineers interested in the application of equipment of this type. To those desiring to add a knowledge of electronic principles to their understanding of electric-current theory and practice, this book will undoubtedly appeal.

The volume is confined closely to the explanation of electronic principles as applied to industrial equipment. Discussions of complicated phenomena, such as that of the oscillator, are confined to basic considerations with references supplied for supplementary reading. Among the electronic applications treated are: High-frequency heating, motor controls, timing, and regulation.

Discussion of x-ray technique and application recently published by the Eastman Kodak Co. is directed to both the professional radiographer and the engineer interested in learning about the possibilities of the science. Coverage includes a description of the process and apparatus as well as detailed information concerning radiographic techniques. Among the subjects discussed are: Geometric principles, exposure factors, radiographic screens, special techniques and processing. Numerous tables and curves augment the general treatment and three appendixes cover sensitometric characteristics, protection and trouble shooting. Entitled, "Radiography in Modern Industry," the paperbound volume measures 8½ by 11 inches and has 122 pages. It is available at $3.00 from Eastman Kodak Co., Rochester 4, N. Y.



Introducing... PAN HEAD

Newest Member of the CLUTCH HEAD *Family*

This is the new style head approved by the American Standard Association... as shown in their Publication, A. S. A. B18.6—1947

America's Most Modern Screw now makes Pan Head with the Clutch Recess available in all standard sizes . . . a full-ranking member of the CLUTCH HEAD family of Machine Screws, Sheet Metal and Thread-forming Screws.

The A.S.A. establishment of the Pan Head as a standard style works in the interest of economy to the user. Experience to date indicates that Pan Head will eventually supersede Round Heads and possibly other style heads . . . with a resultant simplification of inventory by reducing the variety of screws carried in stock.

Since its adaptation to the Clutch Recess, the versatility of the Pan Head, in structure and finished appearance, has met with such popular acceptance by CLUTCH HEAD users that we recommend its investigation from the angles of economy and greater efficiency.

The Pan Head lends itself in a high degree of adaptability to the special features of the Clutch Recess . . . features not matched by any other screw and which have won for CLUTCH HEAD recognition as "America's Most Modern Screw."

SEND FOR SAMPLES

Your request will bring you package assortment of screws, including Pan Head; sample Type "A" Bit and illustrated Brochure. At your own desk, you can check CLUTCH HEAD'S *exclusive features for safer, faster, and easier driving... accorded credit for stepping up assembly productions 15% to 50%.*

CLUTCH HEAD SCREWS

UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

MEN...of machines

A. J. Raitch

A. J. RAITCH has been appointed chief engineer to be in charge of all the Le Roi company's tractor and mower engineering at its Centaur Division, Greenwich, O. An engineering graduate of Detroit University, Mr. Raitch has been active in industrial engineering for the past seventeen years. An authority on tractor engineering, since 1931 he has served as chief engineer of the tractor divisions of Plymouth Locomotive Co., and Graham-Paige Corp., also as chief engineer of the Bolens Products Co., Port Washington, Wis., manufacturer of small garden tractors and implements. During the war, as an officer in the Navy Reserve, Mr. Raitch was an assistant inspector of naval materials at San Francisco.

FRANK W. FINK has been appointed chief engineer of the San Diego division of Consolidated Vultee Aircraft Corp., the position formerly held by B. W. SHEAHAN. Mr. Fink joined Consolidated in 1935, serving first as design engineer in the power plant group and transferring later to the aerodynamic section. During the war he was project engineer on the XB-24 bomber and later advanced to chief production engineer. Immediately following the war he was appointed chief engineer for airline and military airplanes, the position he held until his recent promotion. Mr. Fink is a mechanical engineering graduate of the University of Colorado, 1928, and holds memberships in the Institute of Aeronautical Sciences and the Society of Automotive Engineers, recently being appointed to the SAE administrative board.

Frank W. Fink

LOUIS J. BAUDIS, associated with the Bullard Co. since 1935, has recently been appointed its chief engineer. Awarded the David Orr prize as the ranking engineer among those graduating in 1935 from New York University, Mr. Baudis joined the machine tool company immediately following his graduation. For two years he served in various of the company's production departments and in 1937 was appointed to the standards department. He remained there until 1943 when he was appointed assistant to E. P. Bullard III, vice president in charge of manufacturing. Throughout the war Mr. Baudis assisted Mr. Bullard on production and engineering, and prior to his recent advancement was development engineer and service manager.

Louis J. Baudis

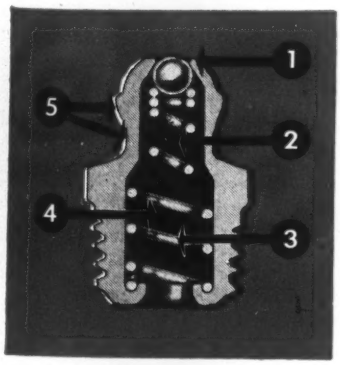
•

THEODORE P. WRIGHT, who recently resigned as head of the Civil Aeronautics Administration, has been appointed president, Cornell Research Foundation, and vice president, Cornell University. Mr. Wright has been active in the aviation industry since 1918 when he enrolled as an ensign in the Naval Reserve Flying Corps following graduation from Knox College and Massachusetts Institute of Technology. Later he joined Curtiss Aeroplane & Motor Co., continuing in key executive capacities after it became Curtiss-Wright Corp. Mr.

New Improved KLEENSEAL Fittings

- KEEP DIRT OUT OF BEARINGS...
- REQUIRE LESS PRESSURE BY 20%
- PROVIDE MORE LUBRICANT VOLUME BY 30%
- SECURELY CONTACTED BY ALL COMMERCIAL HYDRAULIC COUPLERS

These Grease Fittings (Bullneck Type), were originally developed by the Lincoln Engineering Company for the United States Army Ordnance Department. They incorporate the most desirable characteristics of former types of high-pressure fittings and permit standardization on one, all-purpose fitting (Bullneck Type). They represent the first major improvement in grease fitting design that does not obsolete couplers and grease guns in general use today—they can be securely contacted by all commercial hydraulic couplers.



Check THESE FEATURES

1. . . . **Flush Ball Check**—Head can be wiped clean without forcing dirt into fitting and into bearing.
2. . . . **Large Grease Passage**—The larger internal diameter permits greater flow of lubricant.
3. . . . **Special Spring Design**—Spring cannot be compressed to restrict flow of the heaviest lubricants.
4. . . . **Greater Flow—Less Resistance**—The larger internal area requires less pressure to clear the fitting.
5. . . . **Sturdier Construction**—Enlarged neck size and projecting locking pad protect against damage and wear.

Trade names LINCOLN, KLEENSEAL, BULLNECK registered U. S. Pat. Office.

LINCOLN

Leaders in Lubricating Equipment for a Quarter Century

- APPLY THE RIGHT LUBRICANT
- IN THE RIGHT QUANTITY
- AT THE RIGHT TIME

Lincoln Engineering Company
5701 Natural Bridge Avenue, St. Louis 20, Mo., U. S. A.

Gentlemen:

I would like to have literature and engineering data on the new improved Kleenseal Fittings (Bullneck Type).

NAME_____

TITLE_____

COMPANY_____

ADDRESS_____

CITY_____ STATE_____

MD 848-7A

USE THE NYLON
HEYCO
STRAIN RELIEF BUSHING
to anchor the cord set to your electrically operated
MACHINES, EQUIPMENT & APPLIANCES

TWO STANDARD HEYCO TYPES
. . . for all wire sizes

UL RECOGNITION SERVICE

● The Nylon Heyco is an insulating grommet that is snapped into a hole in the chassis of a product. It anchors the cord set firmly to the chassis and imparts a positive non-slip grip—it does not injure the wire. Product life is greatly increased by preventing all strain on terminal connections as well as preventing cord wear at chassis entrance.

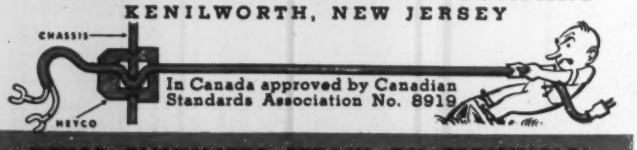
IT'S EASY TO APPLY HEYCO
1. Slip HEYCO on wire.
2. Snap HEYCO into hole.

Here's what Heyco does-

1. Absorbs cord pull, push and torque
2. Insulates wire from chassis
3. Prevents wire from fraying
4. Eliminates tying wire knots
5. Improves product appearance at low cost

TEST SAMPLES WILL BE SENT ON RECEIPT OF WIRE SPECIFICATIONS

HEYMAN MANUFACTURING COMPANY
KENILWORTH, NEW JERSEY



In Canada approved by Canadian Standards Association No. 8919

HEYCO ELIMINATES STRAIN ON TERMINALS!

Wright developed the "project engineer" system of airplane construction as well as pioneering in other aviation developments.

♦

DR. W. E. WILSON has been appointed director of research of the hydraulic division, Sundstrand Machine Tool Co. Prior to joining Sundstrand Dr. Wilson was active as a consulting engineer and during the war was chairman of the department of fluid mechanics, Armour Research Foundation, and directed its numerous research programs on positive displacement pumps and motors. Previously he had been head of the department of mechanics and hydraulics, Colorado School of Mines where he had taught hydraulics and mechanics for eight years.

♦

H. G. BAUER, for the past two years manager of the marine division, De Laval Steam Turbine Co., has been appointed executive engineer. Mr. Bauer joined the De Laval organization in 1937 after extensive marine engineering and shipbuilding experience in Europe and in this country.

♦

PAUL H. MAURER returns to Redmond Co. Inc., after a three-year absence to assume the newly created post of director of engineering in which he will be in charge of all engineering work including research, development and product engineering. For the past three years Mr. Maurer was associated with National Pneumatic Co. as manager of industrial sales and later as chief engineer.

♦

ROBERT E. BUSEY will serve as acting chief engineer, Willys-Overland Motors, reporting to DELMAR G. ROOS, vice president in charge of engineering. WALTER D. APPEL, fomerly chief engineer, recently was named director of puchasing.

♦

CHRIS A. WIEPKING, research engineer, Delta Mfg. Division, Rockwell Mfg. Co., recently has been named chief engineer.

♦

DR. ALBERT W. HULL, assistant director of General Electric Co. research laboratory, and DR. RICHARD W. PORTER, special project engineer in the marine and aeronautics engineering division of the company were among the thirteen scientists to receive honorary degrees from Yale university recently. Both were awarded the degree of Doctor of Science.

♦

GEORGE W. VON HOFE, president of the New Jersey Machine Corp., Hoboken, N. J., has been re-elected president of the Packaging Machinery Manufacturers Institute.

♦

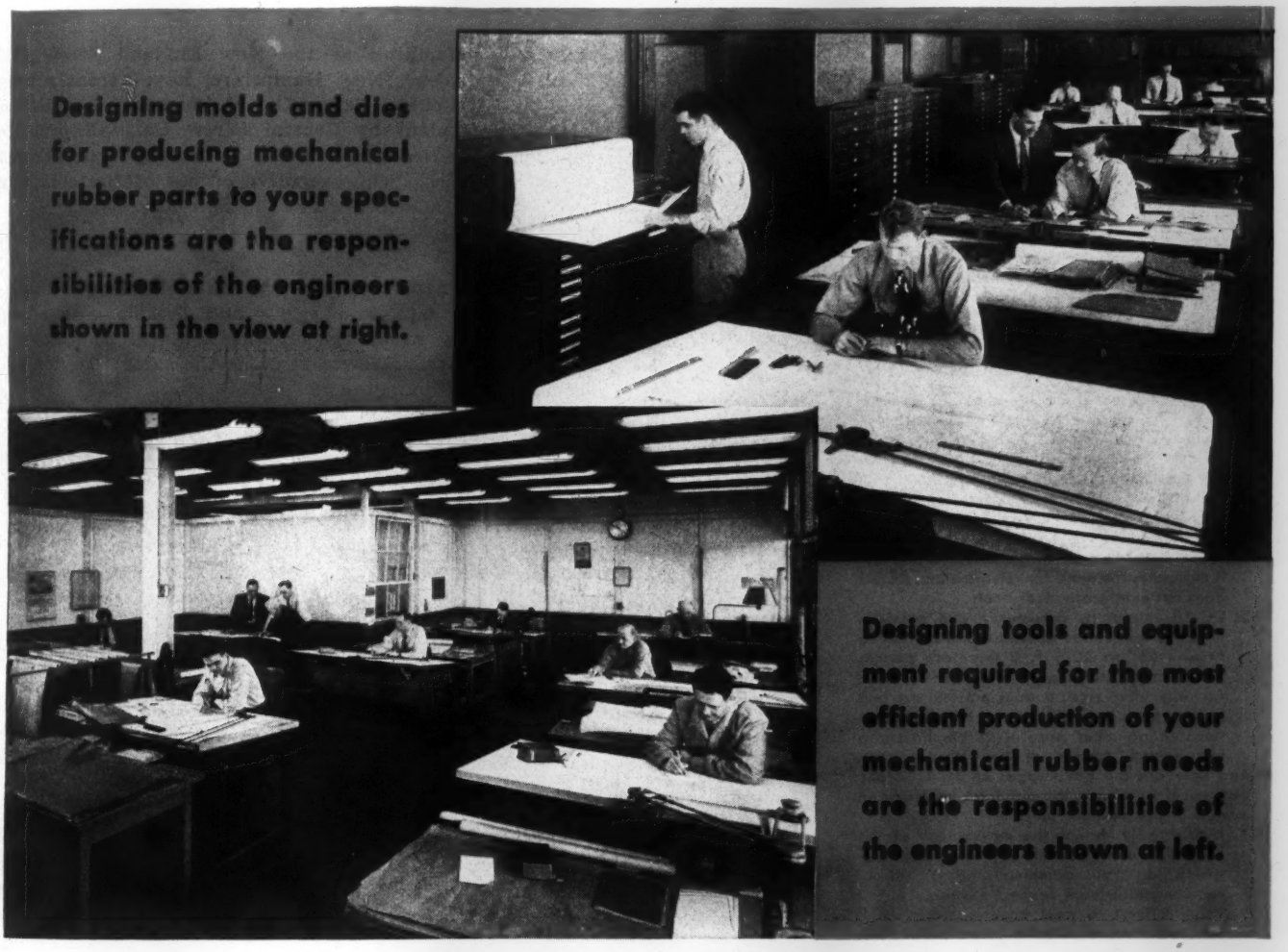
LE VERNE VERZIER has joined Lupomatic Industries Inc., New York, as director of development.

♦

RICHARD SCOTT HUXTABLE, assistant to George W. Codrington, vice president and general manager of the Clevland Diesel Engine division of General Motors Corp., has been appointed executive vice president and general manager, Fawick Airflex Co. Inc., Cleveland. Mr. Huxtable had been connected with the Cleveland Diesel Engine division since 1934.

How to Add Complete Rubber Engineering Facilities to Your Plant

[WITHOUT ONE CENT OF INVESTMENT BY YOUR COMPANY]



Designing molds and dies for producing mechanical rubber parts to your specifications are the responsibilities of the engineers shown in the view at right.

Designing tools and equipment required for the most efficient production of your mechanical rubber needs are the responsibilities of the engineers shown at left.

● When you refer your mechanical molded, extruded, or bonded rubber parts requirements to The Ohio Rubber Company it is almost like adding complete rubber engineering facilities to your plant. Our two specialized staffs of engineers are prepared to coordinate their skills with your engineers to attain the most efficient solutions to your problems in rubber. Suggestion: Ask us to have our representative in your area discuss details of YOUR specific requirements.

This is the 2nd of a series of messages relating to:

THE OHIO RUBBER COMPANY

WILLOUGHBY, OHIO

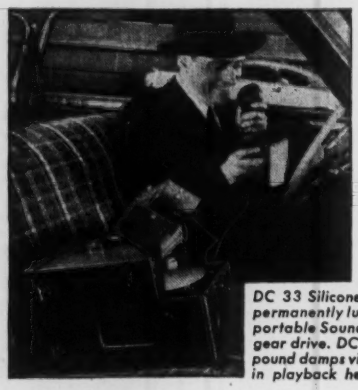
OHIO RUBBER CO

FACTORIES: WILLOUGHBY, OHIO • LONG BEACH, CALIF. • CONNEAUTVILLE, PA.
BRANCH OFFICES: DETROIT • NEW YORK • CHICAGO • INDIANAPOLIS • CLEVELAND • BOSTON

Silicone News

DC

DC Silicones Help Maintain High Performance Standards of Portable SoundScriber



DC 33 Silicone Grease permanently lubricates portable SoundScriber gear drive. DC 4 Compound damps vibrations in playback head.

A portable recording machine that can be used in your car is obviously subject to operation in extremes of temperature. Yet turntable speeds are critical. The SoundScriber Corporation, New Haven, Connecticut, has solved this problem with a sealed gear drive mechanism permanently lubricated with DC 33 Silicone Grease.

Here are the benefits reported from the use of DC 33 Silicone Grease in the drive mechanism:

1. It eliminates the need for relubrication; DC 33 does not evaporate or dry out.
2. DC 33 retains its consistency and provides effective lubrication at temperatures ranging from —95° to 300° F. It therefore permits the motor to start easily and operate satisfactorily when cold; it does not thin out and leak through bearing joints at high temperatures as do other greases or oils having a low enough torque at low temperatures.

And here are the advantages of DC 4 Silicone Compound as used in the playback heads of all models of the SoundScriber:

1. It effectively damps the moving coil of this playback uniformly at any temperature—giving an "oil damping" effect without requiring elaborate sealing devices.
2. DC 4 is inert to most organic materials. It can, therefore, be used in direct contact with the rubber supports in the playback heads.

Dow Corning Silicone Products are notably indifferent to temperature changes. They possess other unique properties that may be even more useful to you. We've had nearly five years' experience in the commercial production of Silicone Greases, Fluids, Resins, and Silastic*. Telephone the branch office nearest you for engineering assistance, or write for Catalog 81-4.

*TRADEMARK FOR DOW CORNING SILICONE RUBBER

DOW CORNING CORPORATION
MIDLAND, MICHIGAN
New York • Chicago • Cleveland • Los Angeles
In Canada: Fiberglas Canada, Ltd., Toronto
In England: Albright and Wilson, Ltd., London

Dow Corning
FIRST IN SILICONES

Dow Corning Silicone Products Include

FLUIDS
Damping
Hydraulic
Dielectric
Waterproofing
Lubricating
Diffusion Pump
Mold Release

GREASES
High Temperature
Low Temperature
Valve Lubricants
Stopcock
High Vacuum

COMPOUNDS
Ignition Sealing
Antifoam A

RESINS
Electrical Insulating
Laminating
Protective Coatings

SILASTIC*
Molding
Extruding
Coating
Luminating

*Trade Mark Dow Corning Corporation

BUSINESS AND SALES BRIEFS

AMONG other appointments recently made by the Allegheny Ludlum Steel Corp. was that of C. M. Binney as assistant district manager of the New York Sales office. Mr. Binney will serve as assistant to B. W. Johnson.

♦

Complete facilities of the New England Screw Co., located at Keene, New Hampshire, have recently been sold to the Central Screw Co. of Chicago. Acquisition of the new plant by Central Screw Co. will promote quicker deliveries of fasteners to heavily concentrated industrial areas. The executive offices of the company will continue their location at Chicago.

♦

Change of name has been announced by Zallea, manufacturers of expansion joints and flexible connectors. New name by which the organization will be known is Zallea Brothers. Address remains the same: Taylor and Locust Sts., Wilmington 99.

♦

Election has been announced of Charles H. Bitzer as president of Harris Products Co. Other changes within the company include election of R. G. Bradley to the post of vice-president in charge of sales and D. T. Bradley to vice-president in charge of engineering. At the same time, C. R. Wight has been named general manager.

♦

J. L. McDermott has recently been named western sales manager of the Glyco-Ryertex Div. of Joseph T. Ryerson & Co. Inc. Mr. McDermott is making his headquarters at the company's Chicago plant.

♦

Completion of a new one-story plant has been announced by the Camcar Products Co. of Rockford, Illinois. New facilities will be largely devoted to manufacturing parts involving cold forging.

♦

National Castings Council, an organization comprised of eight national foundry associations, has recently been formed with Howard A. Stockwell as president. Other officers are: Otto A. Pfaff, vice president; F. Ray Fleig, treasurer; and Frank G. Steinebach, secretary. Office of the secretary is 918 Penton Bldg., Cleveland 13.

♦

According to a recent announcement, The Carpenter Steel Co. has placed Norman C. Einwechter in charge of its Philadelphia-Reading sales territory. Mr. Einwechter, who is assistant to the vice president of the company, will have his headquarters in Philadelphia.

♦

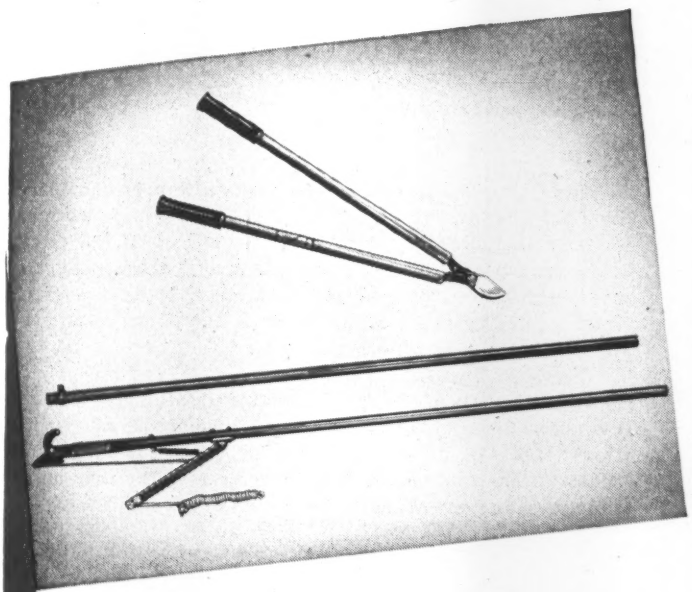
Purchase of the induction heating division of The Budd Co. has been made by the Tocco division of the Ohio Crankshaft Co. Purchase includes machinery,

Take a leaf from this revealing

new book!

Here's the inside story on how manufacturers of a wide range of products—from parts for aircraft to portable tools—have found that it pays to use magnesium. It's a book of actual case studies of successful magnesium applications—a book that will be of direct interest to you.

MANUALLY HANDLED EQUIPMENT



PRUNING SHEARS, HEDGE SHEARS AND TREE TRIMMERS

PROBLEM

The breaking of a wooden handle while operating a pair of pruning shears could mean serious personal injury. A far-sighted manufacturer began to look for a material that was more durable yet possessed the necessary strength. In addition it was important that the new material be in plentiful supply since his search was prompted by the scarcity of high grade white ash and hickory. It was also desirable to lighten the over-all weight of the pruning shears.

SOLUTION

During the past two years the manufacturer has been using magnesium handles on five different products and there have been no instances of breakage. Since magnesium is not subject to warping or splintering and has an excellent combination of light weight, strength and stiffness, it is ideally suited to this application. The slight additional cost of these magnesium handles is more than offset by these advantages. Magnesium extruded tubing is simple to fabricate and has only to be riveted to the pruner head to make a firm attachment. Adding an enamel coating and a pair of hand grips cemented on the ends completes the finished product.

You, too, can make Magnesium Pay!

Magnesium pays for others—it can pay for you. Get the facts—send for this new study today.

Magnesium Division
The Dow Chemical Company • Midland, Michigan

Send me the study MP 26 "How Magnesium Pays." I understand there is no obligation to me.

Name_____

Title_____

Firm_____

Address_____

THE DOW CHEMICAL COMPANY • MIDLAND, MICHIGAN
New York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago
St. Louis • Houston • San Francisco • Los Angeles • Seattle
Dow Chemical of Canada, Limited, Toronto, Canada

Found in the BEST PRODUCTS

• Any product is only as good as its fastenings. The best-made machines and equipment always employ fastenings that are as capable and efficient as the products of which they are a vital part.

Chicago "Safety Plus" Socket Head Screws are used in many of the finest screw-fastened products manufactured today. They improve these products by providing the greatest fastening strength with the least possible weight.

"Safety Plus" fastenings are made of the highest quality materials and manufactured under the most modern processes. They are close-tolerance tested throughout every step of production to assure absolute perfection in thread and dimension. You'll find them superior in every way for your product.

Chicago "SAFETY PLUS" line includes:

Socket Head Cap Screws • Socket Set Screws • Stripper Bolts • Square Head Dog Point Set Screws • Socket Pipe Plugs • Keys for "Safety Plus" Products.

Complete line includes:

Hexagon Head Cap Screws • Square Head Cup Point Set Screws • Headless Set Screws • Fillister Head Cap Screws • Flat Head Cap Screws • Taper Pins • Milled Studs • Semi-Finished Hexagon Nuts • Semi-Finished Hexagon Castellated Nuts.

These Fine Products are sold only thru Authorized Distributors

1872 75 1948

THE CHICAGO SCREW CO.
ESTABLISHED 1872
1026 SO. HOMAN AVENUE CHICAGO 24, ILL.

equipment, inventory and patents, but not the land or the buildings.

♦

A new sales territory has been formed by Kurz-Kasch, Inc. of Dayton, Ohio. Comprising western New York, western Pennsylvania and northeastern Ohio, the territory will be under Charles F. Hamilton, who will maintain headquarters in the News-Herald Bldg., Franklin, Pa.

♦

Peter F. Hurst, formerly executive vice president and general manager of Aeroquip Corp., has been elected president of the organization. Mr. Hurst was instrumental in the organization of the corporation in 1940.

♦

Connected with the National Screw and Mfg. Co. in the capacity of executive vice president and director, Charles F. Newpher has recently been elected to the board of directors of the Lock Thread Corp. Mr. Newpher, having 35 years of experience with fasteners, is a leading authority in the field.

♦

Three changes in sales personnel have been announced by SKF Industries, Inc.: John H. Tipton is now assistant district manager of the Cincinnati office; M. H. Courtenay has been appointed assistant district manager of the Atlanta office; and E. A. Hutson is field engineer in the railway sales department of the Chicago office.

♦

Appointment of Scott D. Baumer, formerly assistant manager of the technical sales division of Air Reduction Sales Co., to managership of that division has recently been announced. Succeeding Mr. Baumer as assistant manager of sales is Edward H. Roper.

♦

Formerly in charge of centrifugal compressor sales of the De Laval Steam Turbine Co., J. P. Stewart has been appointed manager of commercial and marine sales.

♦

The Elmira Foundry Co. Inc., formerly a General Electric affiliate, has been made an integral part of the parent company. The foundry becomes one of nine major works and 24 factories composing the General Electric apparatus department.

♦

Connected with the Trabon Engineering Corp. since 1936, Wilbur Deutsch has been appointed general sales manager. Mr. Deutsch was formerly in charge of sales and service for the company's west coast territory.

♦

New officers have recently been elected by the Brass and Bronze Ingot Institute. President is Leo Halpern of Federated Metals Div. of American Smelting and Refining Co. Marvin M. Cohn, of L. A. Cohn & Bros., Inc. is vice-president; and Howard K. Chapman of H. Kramer & Co. is treasurer of the institute.

♦

According to a recent announcement, Sundstrand Machine Tool Co. has appointed two new sales representatives. In California, the Burgan Machinery Co., 9527 5th Ave., Inglewood has been named western representative;

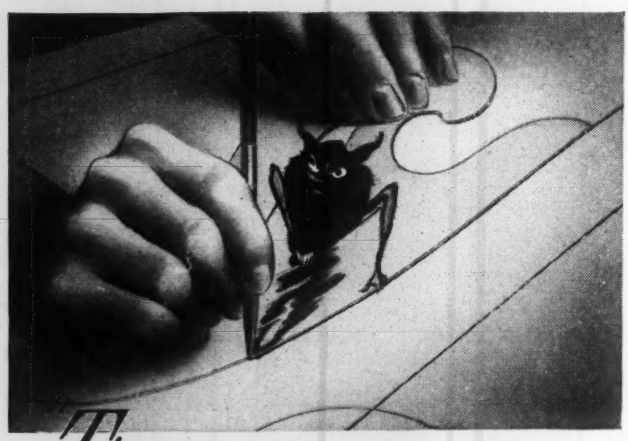
Save machining time, produce accurate parts

With J&L Precision Ground Cold Finished Steel

The size accuracy and surface finish of J&L Precision Ground Cold Finished Steel is used to advantage by many manufacturers in machining parts. They find it eliminates part of the machining operation, saves tool wear and tool changes. J&L Precision Ground is available in a wide range of grades in sizes ¼" and larger from mill or your nearest J&L warehouse or distributor.

JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PA.



They used to call me 'THE CREEP'..."

". . . but that was before the boss discovered there was a *difference* in tracing cloths. Ink just can't help creeping and feathering when you use a French Curve, if the tracing cloth has a poor surface."

Arkwright gives you the kind of surface that practically assures razor-sharp lines. Oil, wax and soap-free mechanical processing assures uniform capillarity. Even the weaving and bleaching of special cloth is part of Arkwright's exhaustive system of standards, tests and inspections—to *prevent* pinholes, thick threads and the many other things that cause spoiled tracings, lost time.

Why not try Arkwright and see for yourself what a difference there is? Send for free working samples. Arkwright Finishing Company, Providence, R. I.

All Arkwright Tracing Cloths have these 6 important advantages

1 *Erasures re-ink without "feathering" or "creeping"*
2 *Prints are always sharp and clean*
3 *Tracings never discolor or become brittle*
4 *No surface oils, soaps or waxes to dry out*
5 *No pinholes or thick threads.*
6 *Mechanical processing creates permanent transparency*

Arkwright
TRACING CLOTHS
AMERICA'S STANDARD FOR OVER 25 YEARS

while in Connecticut, P. A. Anderson, located at Box 166 Lakeville, is the new eastern representative.

♦

According to a recent announcement, DuMont Electric Corp. has moved its factory and offices to a new building. New location is 302-8 Dyckman St., New York 34.

♦

Election of Walter E. Remmers as president of Electro Metallurgical Co. has been announced by the board of directors. Mr. Remmers was formerly vice-president and director of the company.

♦

Selenium Corp. of America, an affiliate of Vickers Inc. and a unit of the Sperry Corp., has been dissolved and succeeded by Vickers Electric Div., Vickers Inc. The address is 2160 E. Imperial Highway, El Segundo, Calif.

♦

Appointment of Richard H. Diesel as assistant general sales manager of the Stamford Div. of The Yale & Towne Mfg. Co. has been announced. Mr. Diesel had been manager of aircraft and automotive sales, with headquarters in Detroit.

♦

Field engineering activities for Federal-Mogul Service will now be headed by Walter E. Thill. Mr. Thill will supervise preparation of technical service publications, field service meetings and service engineering studies.

♦

Election has been announced of F. E. Ericson as president of The Barden Corp., manufacturers of high-precision ball bearings. Mr. Ericson has been vice president and general manager of the corporation since its organization.

♦

With offices at 820 S. Maple Ave., Oak Park, Ill., Richard H. Khuen is now serving as regional sales representative of the Wel-Met Co. Mr. Khuen has the territory covering Illinois, Iowa, Wisconsin and Missouri.

♦

L. F. McCaffrey, special engineer for the metal industries in the Allis-Chalmers industrial sales department is now located in the organization's Pittsburgh district office. Mr. McCaffrey's services will continue to be available to all of the company's offices.

♦

Establishment of Dalc Bearings Inc. with offices at 1974 Broadway, New York, has been announced. Specializing in the sale of antifriction bearings, the new organization is headed by Clayton A. Decker, president.

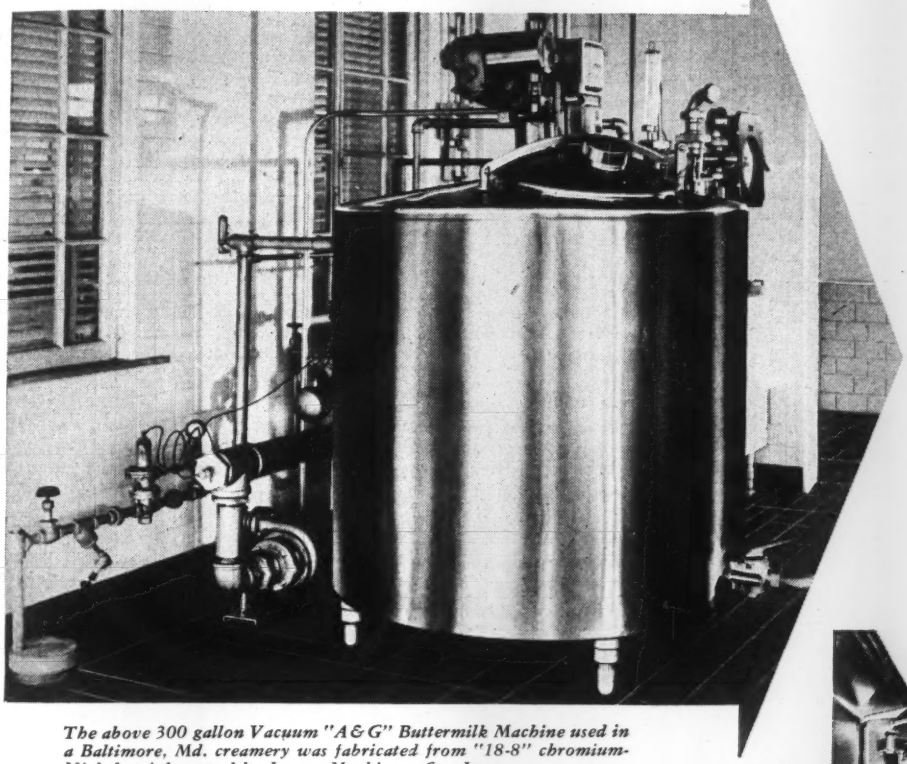
♦

Formerly with the Superior Engine Div. of the National Supply Co., Lloyd A. Harris, diesel engine specialist, will now represent the company's plant products division. He will specialize in the sale and service of Superior engines.

♦

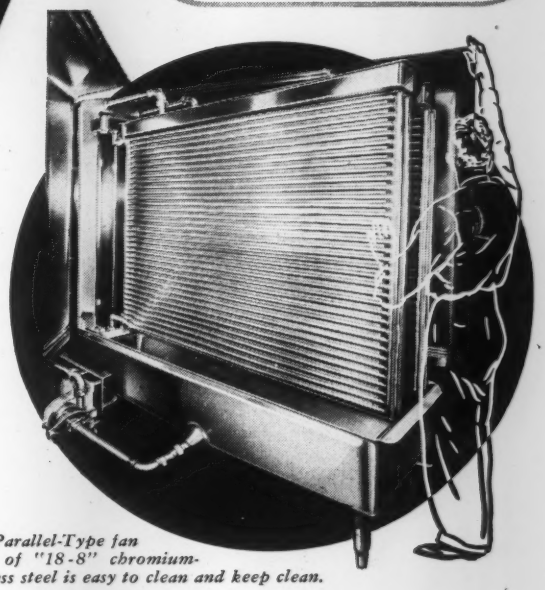
According to a recent announcement, the Providence Custom Molding Co. Inc. has been organized to injection mold all thermoplastic materials. The company, located at 690 Central Ave., Pawtucket, R. I., also offers consulting and assembly service.

When Product Purity is Essential



The above 300 gallon Vacuum "A & G" Buttermilk Machine used in a Baltimore, Md. creamery was fabricated from "18-8" chromium-Nickel stainless steel by Jensen Machinery Co., Inc.

...take a tip from Dairymen



This Jensen Parallel-Type fan cooler made of "18-8" chromium-Nickel stainless steel is easy to clean and keep clean.

Even the *least* contamination by bacteria spells D-A-N-G-E-R to processors of milk.

The faintest contamination from products of corrosion may affect the taste.

And that's why chromium Nickel austenitic stainless steel is used for so many pasteurizers, storage tanks, surface coolers, piping and other units for the dairy products industry.

Easy to keep hygienically clean, standard "18-8" stainless is highly resistant to foodstuffs, atmosphere, most organic and a great many inorganic chemicals, and to dyes and sterilizing solutions.

In addition, for many applications in various fields where resistance to corrosion, impact, wear and abrasion are required, "18-8" permits cutting bulk and dead weight without sacrificing strength or durability. Its great resistance to both high and low temperature effects is especially valuable.

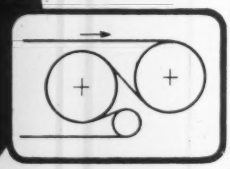
Leading steel companies produce stainless steel containing Nickel in sheet, strip, bar and tubular form. A list of the sources of supply will be furnished on request.

EMBLEM OF INCO SERVICE
TRADE MARK

Over the years, International Nickel has accumulated a fund of useful information on the selection, fabrication, treatment and performance of engineering alloy steels, stainless steels, cast irons, copper-base and other alloys containing Nickel. This information is yours for the asking. Write for "List A" of available publications.

THE INTERNATIONAL NICKEL COMPANY, INC. 67 WALL STREET NEW YORK 5, N. Y.

When Production Depends on Product Movement



Large or small, for tough or simple installations, there's a Cambridge belt to do the job. Above: Sintering powdered metal parts in a Cambridge tray on a Cambridge Rod Reinforced Belt.

You Can Depend on a *Cambridge* WIRE BELT

If your products or raw materials must move during processing, call in a Cambridge engineer to discuss the proper woven wire belt for your conveyor machines.

Let us show you how Cambridge Engineered belts are specially built for specific installations—specially built from any metal or alloy to any mesh or weave.

Or, for information on the entire Cambridge wire belt line, write for your *FREE* copy of this illustrated 130 page booklet.

There's a Cambridge Sales Engineering Office near you

CAMBRIDGE WIRE CLOTH CO.

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BOSTON • NEW YORK • BALTIMORE • PITTSBURGH
DETROIT • CHICAGO • SAN FRANCISCO • ST. LOUIS

MEETINGS AND EXPOSITIONS

Mar. 30-31, Apr. 1—

Society of Automotive Engineers Inc. National transportation meeting to be held at Hotel Bellevue-Stratford, Philadelphia. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Apr. 5-8—

National Association of Corrosion Engineers. Fourth annual conference and exhibition to be held at the Jefferson Hotel, St. Louis, Mo. W. Z. Friend, 67 Wall St., New York 5, is chairman of the publicity committee.

Apr. 5-8—

Southern Machinery and Metals Exposition Inc. Third annual exposition to be held at Atlanta Municipal Auditorium, Atlanta, Ga. Michael F. Wiedl, Jr., 267 E. Paces Ferry Rd., Atlanta 5, is managing director.

Apr. 13-15—

Society of Automotive Engineers Inc. National aeronautic and air transport meeting to be held at Hotel New Yorker, New York. John A. C. Warner, 29 West 39th St., New York 18, is secretary and general manager.

Apr. 15-16—

Metal Powder Association. Fourth annual spring meeting and exhibit to be held at the Drake Hotel, Chicago. Additional information may be obtained from the headquarters of the association, 420 Lexington Ave., New York 17.

Apr. 19-21—

American Society of Lubrication Engineers. Third annual convention and exhibit to be held at the Hotel Statler, Buffalo. W. F. Leonard, 343 South Dearborn St., Chicago 4, is secretary-treasurer.

Apr. 26-27—

Association of Iron and Steel Engineers. Annual spring conference to be held at Buffalo. Additional information may be obtained from society headquarters, 1010 Empire Building, Pittsburgh 22, Pa. T. J. Ess is managing director.

Apr. 26-30—

American Management Association. Seventeenth packaging exposition and conference to be held at Cleveland Public Auditorium, Cleveland. Clapp & Poliak Inc., 350 Fifth Avenue, New York 1, is exposition management.

May 3-7—

American Foundrymen's Association. Annual convention and exhibit to be held in Philadelphia Convention Hall, Philadelphia. William W. Maloney, 222 W. Adams St., Chicago 6, is secretary-treasurer.

May 20-21—

Society of the Plastics Industry. Annual meeting to be held at Atlantic City, N. J. Additional information may be obtained from the headquarters of the society, 295 Madison Ave., New York 17. William T. Cruse is executive vice president.

May 20-22—

American Society of Mechanical Engineers. Oil and Gas Power division meeting to be held in St. Louis, Mo. Additional information may be obtained from society headquarters at 29 West 39th St., New York 18. C. E. Davies is secretary.

May 27-29—

Society for Experimental Stress Analysis. Spring meeting to be held at the Roosevelt Hotel, Pittsburgh, Pa. W. M. Murray, P. O. Box 168, Cambridge 39, Mass., is secretary-treasurer.

May 30-June 4—

American Society of Mechanical Engineers. Semiannual meeting to be held at Milwaukee, Wis. Additional information may be obtained from society headquarters at 29 West 39th St., New York 18. C. E. Davies is secretary.

NATIONAL OIL SEAL LOGBOOK

WHEN TO SPECIFY SPRINGLESS OIL SEALS

Springless oil seals are recommended only under extremely limited mechanical design conditions, or when great economy in original production is demanded.

STEEL-ENCASED LEATHER

Springless seals are generally of two broad types. First and most common of these is the steel-encased leather seal. (See Fig. 1.) Such seals are often used where complete sealing is not required, and slight leakage is not objectionable. Springless leather seals are also often specified in applications where heavy grease is to be withheld; where temperatures are low and speeds do not exceed 1000 rpm.

STEEL-ENCASED SYNTHETICS

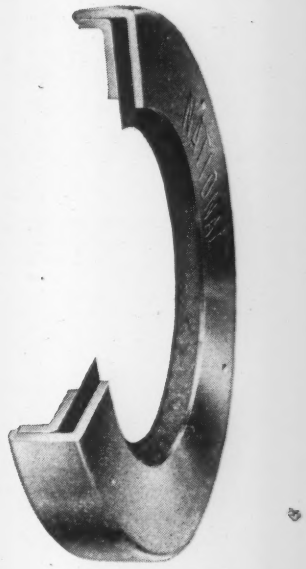
Second and most versatile of the springless designs is the steel-encased synthetic seal. Three different designs, each offering maximum efficiency for a specific application, are shown in Figures 2, 3, 4. Generally speaking, all these designs are superior to steel-encased leather seals. Available for a wide range of applications, they retain lighter oils, and withstand greater temperatures and higher speeds than other springless designs.

MOST COMMON DESIGN

The most common design of these springless synthetics is shown in Figure 2. This type is often specified for wheel applications which are well designed and have good sealing surfaces. This seal is capable of medium pressure, but with slight re-designing can withstand pressure up to 70-100 pounds psi.

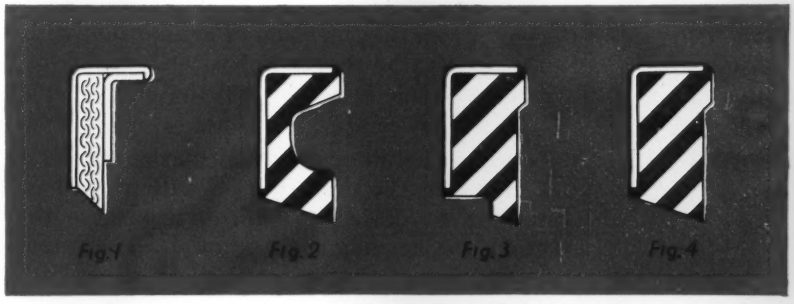
The design shown in Figure 3, is intended primarily for use where temperatures and speeds are high, but eccentricity and runout are low and running parts are accurately made. This seal is limited, however, to not over 5 to 10 pounds psi pressure, and is a compromise seal when space limitations forbid using a spring-loaded type seal.

The seal shown in Fig. 4 is primarily a pressure seal. It has been tested successfully up to 600 pounds



psi fluctuating pressure, at minimum speeds. It is also suggested for extremely small cross-sections of I.D. vs. O.D., where a relief design is not possible. This seal is often specified for use on shafts of $\frac{3}{8}$" diameter and below.

However, before specifying *any* springless oil seal, it is recommended that designers seek the assistance of National Oil Seal Engineers. In most applications the spring-loaded oil seal is much more desirable, and will provide far greater performance.



NATIONAL
OIL AND FLUID SEALS ®

NATIONAL MOTOR BEARING CO., INC.

General Offices: Redwood City, California
Plants: Redwood City and Los Angeles, California; Van Wert, Ohio

CALL IN A NATIONAL ENGINEER FOR RECOMMENDATIONS

BUFFALO: 56 Arlington Place, Grant 2280. **CHICAGO:** Room 2014 Field Building, Central 8663.
CLEVELAND: 210 Heights Rockefeller Building, Yellowstone 2720. **DALLAS:** 30½ Highland Park Village, Justin 8-8453.
DETROIT: Room 1026 Fisher Building, Trinity 1-6363. **HOUSTON:** 6731 Harrisburg Boulevard, Wayside 2621.
LOS ANGELES: 2244 East 37th Street, Kimball 6384. **MILWAUKEE:** 1717 E. Kane Place, Lakeside 2838.
NEW YORK CITY: 122 East 42nd Street, Lexington 2-8260. **PHILADELPHIA:** 401 North Broad Street, Bell-Wainut 2-6997.
REDWOOD CITY, CALIF.: Broadway and National, Emerson 6-3861. **SPRINGFIELD 3, MASSACHUSETTS:** 124 State Street, Springfield 2-1881.
WICHITA: 340 North St. Francis Street, Wichita 2-6971.

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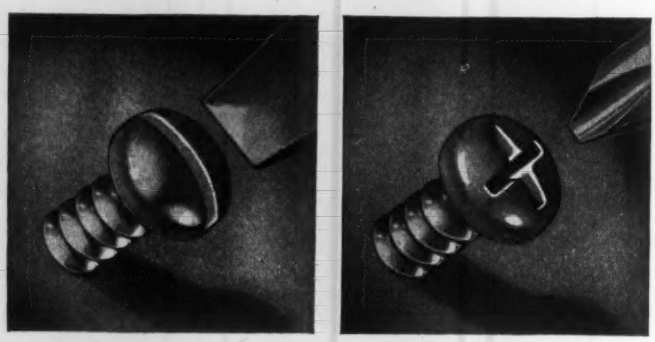
BIGGEST NEWS
IN THE FASTENING FIELD!



ALCOA ALUMINUM
SHEET METAL SCREWS

Tough, corrosion-resistant, slotted or Phillips-head. Type A—gimlet point, or type Z, blunt point. In lengths ranging up to 2″, and in diameters up to No. 14.

For strong, tight, corrosion-free joining of aluminum members—write *Alcoa Aluminum Fasteners* right on your drawings! ALUMINUM COMPANY OF AMERICA, 617 Gulf Building, Pittsburgh 19, Pennsylvania.

FASTEN ALUMINUM
WITH ALUMINUM

Ring Deflection

(*Continued from Page* 134)

by applying the principle of least energy, which is a special case of Castigliano's second theorem.

If, for instance, the loading condition indicated in *Fig.* 5 has to be considered, then cutting the ring at A_1 or A_2 and treating M as the redundant reactive moment there results, from the principle of least energy, the equation $\partial U/\partial M = 0$ from which the magnitude of M can be determined. Once this is done the deflection of the point of application of P can be calculated in the manner already indicated.

In this case the maximum deflection will not occur at the point of application of P. It can be found by inserting a hypothetical load P_n to act at an angular distance α from OA. The deflection of P_n is found as before by applying Castigliano's first theorem, $\Delta = \partial U/\partial P_n$. In this way there is obtained an equation in α and by equating the differential of this equation to zero, the angle at which the maximum deflection will occur is obtained, P_n being equated to zero.

Uniform Load Derivation

UNIFORMLY DISTRIBUTED LOAD: If the ring is subjected to a uniformly distributed load acting along the center of twist of the cross-section, then by virtue of symmetry, the moments at A_1 and A_2 are equal, i.e., $M_{x_o} = wR^2$.

The method of deriving the general deflection equation is the same as that indicated in deriving the deflection equation for any point on the ring outlined earlier in this paper. Four equal hypothetical loads P_n are placed on the ring as shown in *Fig.* 6. The deflection at any one of these loads is given by the equation, $\Delta = \partial U/\partial P_n$, P_n being equated to zero.

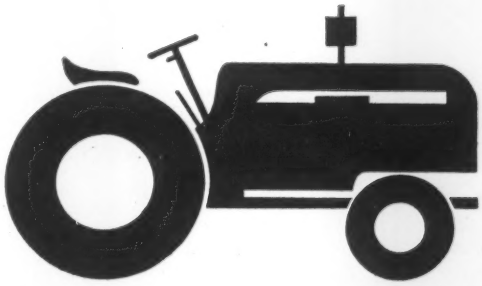
If the distributed load is offset from the center of twist, then, the equivalent loading consists of a uniformly distributed load along the center of twist together with a uniformly distributed couple. If this offset is $\pm\epsilon$, the distributed couple per unit length is $\pm w\epsilon$. From the condition of equilibrium with respect to moments, the moment at A_1 and A_2 due to the distributed couple is $\pm w\epsilon R$ and the effective moment is $M_{x_o} = wR\ (R \pm \epsilon)$.

STRESS DISTRIBUTION: Calculation of the intensity of stress in the cross-section with regard to flexure presents no difficulty. For this purpose the simple theory for bending may be used. The intensity and distribution of the torsional shear stress for sections other than circular is, however, a more complicated matter. However, many such sections have been investigated and some of these have been compiled by A. J. Roark.*

CONCLUSION: By making suitable use of curves presented in *Figs.* 3 and 4 the maximum deflection can be adjusted to design requirements. Invariably the torsional element is by far the greater offender

* Roark, A. J.—*Formulas for Stress and Strain*, McGraw-Hill Book Co., New York, 1938.

DO YOU BUILD TRACTORS?



The manufacturer of a nationally-known farm tractor gets plenty of OSTUCO production-assistance that saves time and money on the assembly line. OSTUCO Tubing and these forming and fabricating operations—flattening, bending, swaging, expanding, threading, angle-cutting, beading and punching—eliminate production line delays and worries!

TRICYCLES?

The frame, stem, forks and hubs of this tricycle are fabricated by OSTUCO. Bending, piercing, flattening, reducing, beading and profiling (cutting the end of the tubing in a half-moon shape to fit exactly into another piece of tubing of like size) are the operations that slash production costs at this toy manufacturer's plant!

Tractors, tools, tricycles . . . transformers, trolley poles and hundreds of other products . . . are being produced more efficiently and economically today with OSTUCO Tubing, formed and fabricated to the most exacting specifications. Almost any product you could name—most likely including your own—can be built in less time, with less effort and at much lower cost because of the experience and skills which OSTUCO makes available to manufacturers in every field of industry.

Here is an opportunity you can't afford to overlook—an opportunity to improve your product, to produce it in less time and make substantial production savings as well. Experienced OSTUCO engineers and skilled OSTUCO craftsmen, working directly from your own blueprints, can create a finished part or product that will meet your individual requirements, greatly simplifying your production problems and helping make yours a better product.

The possibilities of OSTUCO production-assistance are virtually unlimited . . . the complete story is available, without obligation, at the nearest OSTUCO sales office. Write today . . . learn how OSTUCO Tubing can provide time-saving and cost-cutting assistance on your production job.

TOOLS?

Three OSTUCO operations—tapering, expanding and drilling—performed on OSTUCO Tubing, enable manufacturers of rakes, hoes, forks, shovels and other hand tools to turn out more units per man and per machine. There's a real savings in production costs here, and a far better product as well!

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In the United States, Canada, and throughout the world, Briggs & Stratton maintains a vast network of *Authorized Service Stations*, operating under factory supervision and policies.

These stations offer complete engine service — unequalled in the industry — mechanics, factory trained to adjust, repair, or rebuild Briggs & Stratton engines with original parts. One more reason why Briggs & Stratton is "preferred power" by manufacturers, dealers, and users of industrial equipment, farm machinery, and appliances.

BRIGGS & STRATTON CORP., MILWAUKEE 1, WIS. U.S.A.

where the maximum deflection is concerned and this is particularly so when open cross-sections are used. In such cases the torsional rigidity is very small compared with the flexural rigidity. Closed sections should therefore be used whenever possible. The method of analysis is not restricted to rings of uniform cross-sections only, it is also applicable to rings of varying cross-section, but in this case the integration will become relatively more complex.

The foregoing analyses are based on the assumption that all sections of the ring are free to warp. An entirely new element will be introduced if one or more sections are prevented from warping. In this case the torsion will only partially be resisted by shearing stresses in accordance with St. Venant, the remainder being resisted by secondary stresses.

NEW MACHINES

And the Companies Behind Them

Business

ADDING MACHINE. 188 cycles per min. Basic operating cycle rotary rather than reciprocatory; keys and control bar fully automatic; extra add-bar at base of keyboard; power-driven rotary printer. Solderless connections; 1/30-hp, 115-230-volt universal motor. Clary Multiplier Corp., Los Angeles 12.

Checking

GAGING MACHINE. Automatically segregates lapped-finish parts into 27 classifications. Valve plates checked by airflow jets and Sheffield Airlectric gaging heads. Twenty-five receiving stations, two rejection chutes. The Sheffield Corp., Dayton 1, O.

Domestic

ELECTRIC HEATER. Recessed in wall; built-in thermostat control. Heats by radiation and convection. Heating element, nickel-chrome wire on ceramic post. Capacities: 1500, 2000, 3000, and 4000 watts on 230 volts ac. Westinghouse Electric Corp., Pittsburgh 30.

POWER LAWN MOWER. Conventional rotating-blade type; belt and chain driven; 24-in.; equipped with simple hitch for attachment to garden tractor. All-steel construction. The Special Products Div., Lodge & Shipley Co., Cincinnati.

AIR COOLER. Evaporative, window-blower type. Rubber-mounted motor and blower; quickly removable rear louvered panel; trough-type water distributing system; automatic water regulator. Motor, 1/6-hp. Unit cools and filters 1800 cu ft air per min. Palmer Manufacturing Corp., Phoenix, Ariz.

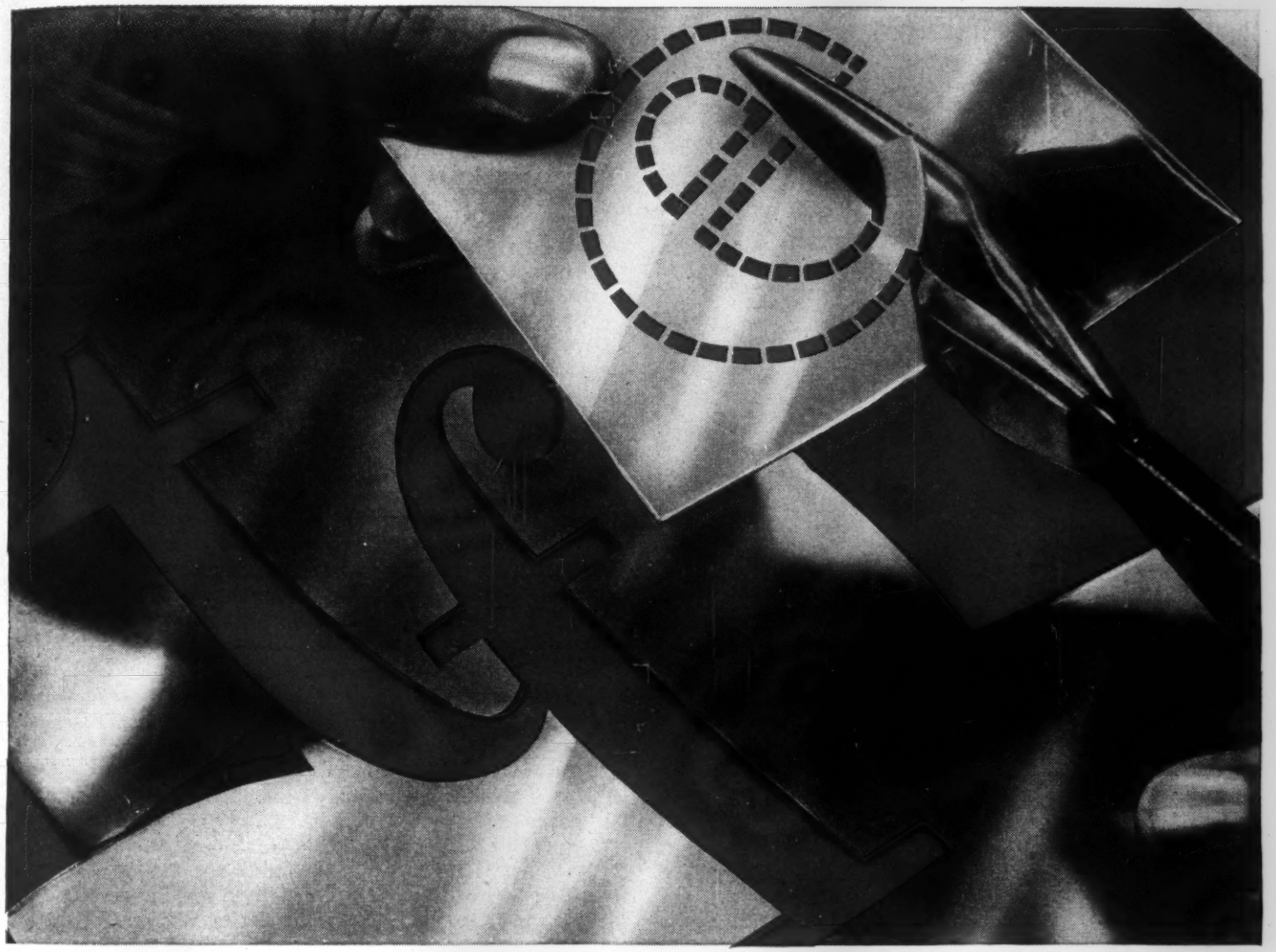
OIL BURNER. Over 90 per cent efficiency. Burns any type oil, less than one gallon per hr. Tip temperature, 2300 F. One moving part—a 1/20 to 1/50-hp motor operating on 110 volts ac-dc. No transformer, electrodes, stack switch, fuel pump or nozzle. Consolidated Industries, Inc., Lafayette, Ind.

Finishing

OVEN CONVEYOR SYSTEM. Conveyor operates through gas-fired oven, carrying painted parts for baking or drying. Ace Conveyor Industries, Inc., Detroit.

Heat Treating

INDUCTION HEATER. High-frequency; self-contained; foot-switch controlled. Twin heating coils, each drawing 125 kw of 960-cycle power. Power source, 700-kw,



t.f.e. it's the only way to cut fastening costs

You pay more money to specify, purchase, stock, inspect, requisition and *use* fasteners than you do to buy them. *True Fastener Economy* means making sure that *every* function involved in the *use* of bolts, nuts, screws, rivets and other fasteners contributes to the desired fastening result—maximum holding power at the lowest possible *total* cost for fastening.

You Get True Fastener Economy When You Cut Costs These Ways

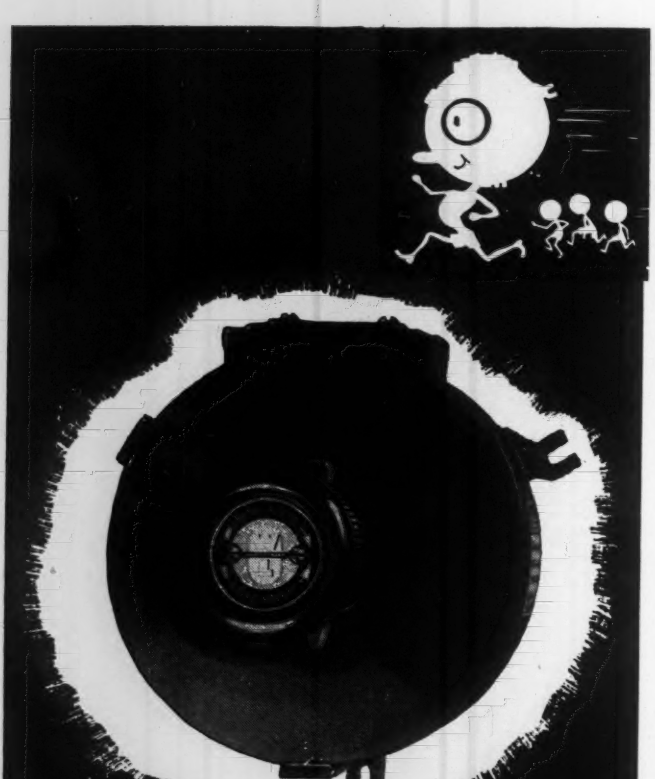
1. Reduce assembly time with accurate, uniform fasteners
2. Make satisfied workers by making assembly work easier
3. Save receiving inspection through supplier's quality control
4. Design assemblies for fewer, stronger fasteners
5. Purchase maximum holding power per dollar of initial cost
6. Lower inventory by standardizing types and sizes of fasteners
7. Simplify purchasing by using one supplier's complete line
8. Improve your product with a quality fastener.

True Fastener Economy is the Lowest Total of All Fastening Costs.

103 Years Making Strong
the Things That Make America Strong

RUSSELL, BURDSALL & WARD BOLT AND NUT COMPANY

Plants at: Port Chester, N. Y., Coraopolis, Pa., Rock Falls, Ill., Los Angeles, Calif. Additional sales offices at: Philadelphia, Detroit, Chicago, Chattanooga, Portland, Seattle. Distributors from coast to coast.



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for less money

WHERE BASIC DESIGNS IN ELECTRIC CONTROLS ARE RESULT-ENGINEERED FOR YOU

Construction of Ward Leonard Vitrohm Rheostats permits use of smaller sizes

A large number of solid rectangular contacts, imbedded in vitreous enamel on a pressed steel base, fit Ward Leonard Rheostats for high watt capacity and long, continued service. Size for size, no other rheostat will perform a similar duty.

Ward Leonard Rheostats are "Result-Engineered". By modifying a basic design, Ward Leonard can often give you the results of a special . . . for the price of a standard.

Write for Rheostat Catalog. Ward Leonard Electric Co., 58 South St., Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

WARD LEONARD
ELECTRIC COMPANY
RESISTORS · RHEOSTATS · RELAYS · CONTROL DEVICES

960-cycle, 400-volt motor-generator. Ajax Electrothermic Corp., Trenton 5, N. J.

Industrial

HUMIDIFIER. Self-contained, humidistat controlled. Four nozzles atomize 35 lb water per hr using 5 cfm atomizing air at 32 psi. Spraying Systems Co., Chicago 24.

VIBRATORY PACKER. Electromagnetic. For tamp-filling small phials, jars, cartons, and cans with powdered or granular materials. Portable; variable power control; no gears, bearings or lubrication. Deck size, 7 x 10 in.; overall height, 4½-in.; power, 110-volt, 60-cycle; input, 9 amps; weight, 13 lb. Syntron Co., Homer City, Pa.

DUST COLLECTOR. Industrial type, self-contained. Over 1300 cfm suction on 6-in. diam inlet with 3-in. water lift. Dust-laden air is cleaned by cyclone separator and spun-glass filter. Floor space, 22 by 53½ in.; overall height, 42 in. Aget-Detroit Co., Ann Arbor, Mich.

HUMIDIFIER. Evaporates over one gallon water per hr. Replaceable spun-glass filter. Walton Laboratories Inc., Irvington 11, N. J.

PORTABLE CRANE. Hydraulic. Enables one man to move ton, positioning load with one hand. Lifting range, 3 ft to 9 ft, 7 in.—with boom extension, 3 ft to 12 ft, 7 in. Lempco Products, Inc., Bedford, O.

Manufacturing

SPOT WELDER. Pivot type, foot operated. Throat depths, 12, 18 and 24 in. Transformer, 10 kva, glass insulated. Both horns adjustable in and out, and rotatable. Passes current only when pressure is applied to work. Fabricated steel construction, prelubricated bearings. Single-phase ac power; 220, 380, 440, or 550 volts; 25, 50 or 60 cycles. Agnew Electric Co., Milford, Mich.

METALLIZING GUN. For metal spraying. New nozzle assures low consumption of acetylene and oxygen; one adjustment for all metallizing wires. Weight, 4 lb, 2 oz. Metallizing Engineering Co. Inc., Long Island City 1, N. Y.

HYDRAULIC PRESS. For straightening weldments, castings, etc. Self-contained; 200-ton. Pressure ram movable over 120 by 38-in. area of bed; ram positioning electrically controlled; dial-type pressure gage. The Hydraulic Press Mfg. Co., Mt. Gilead, O.

PORTABLE REPAIR SHOP. Motorized flexible shaft drives 25 different accessory tools interchangeably. Motor, 1/3 hp, 110-220 volts, 60-cycle. Three speeds: 1400, 2400 and 4500 rpm; V-belt drive. Wyzenbeek & Staff, Inc., Chicago 22.

Materials Handling

STACKING TRUCK. Four models. Capacity, to 4000 lb. Battery-powered through series-wound motors; electro-hydraulic lifting and lowering gear-drive traction unit; safety electric brake; two speeds forward and reverse. Lewis-Shepard Products Inc., Watertown 72, Mass.

PORTABLE CONVEYOR. For all bulk materials. Powered by electric motor or gas engine. Rubber-ply canvas belt; hydraulic lift for positioning; pneumatic tires; automotive type springs; trailer type hitch. Sermat Conveyor Corp., Chicago.

MOTORIZED HAND TRUCK. Equipped with platform or forks; electric or foot lift; dead-man control; double-position brake; capacity, to 6000 lb. Automatic Transportation Co., Chicago 20.

FORK TRUCK. Capacity, 200 lb; 10-hp, twin-cylinder, 4-cycle, air-cooled gasoline engine. Hydraulic lift and tilt; available lift, 7, 8 and 9 ft. Outside turning radius, 59 in. The Buda Co., Harvey, Ill.

Merchandising

ICE CREAM DISPENSER. Self-service; button operated; elec-

trically controlled. Maintains below-zero temperature throughout food compartment; automatic defroster. Frosted Food O'Mat, Inc., Oakland 11, Calif.

Metalworking

DRILL PRESS. Floor type. Capacity, 1½-in. drill. Geared power feed operates in oil reservoir, provides feeds of 0.003, 0.006 and 0.010-in. per spindle revolution. Speed selection by dial. V-belt drive released by spring-loaded lever when changing speeds. Eight spindle speeds; 2-hp motor; electric coolant pump. Sibley Machine & Foundry Corp., South Bend 23, Ind.

DRILL PRESS. Floor type. Spindle speed range, 150-3400 rpm, infinitely variable. Canedy-Otto Mfg. Co., Chicago Heights, Ill.

PORTABLE DRILL PRESS. Powered by detachable No. 2 Moto-Tool having speed of approximately 27,000 rpm. Work brought to drill on 3¼-in. table having 1⅜-in. stroke. Table raised and lowered by rack and pinion. Height, 12 in; length, 6¾ in; width, 4 in; weight, approximately 3½ lb. Dremel Mfg. Co., Racine, Wis.

MILL-ROLL DRESSING MACHINE. Automatic; for steel mills. Standard airless blasting wheel throws chilled iron abrasive against exposed work surface of mill roll. Dresses five rolls per hour. American Wheelabrator & Equipment Corp., Mishawaka, Ind.

SMALL MILLING MACHINE. Two speeds; adjustable stop for cut-offs. Bar stock to 1-in. diam. Mills flats, tangs, squares, Woodruff key slots and keyways. Motor, ¼-hp, 1750 rpm. Hyland Machine Co., Dayton 1, O.

MIXER AND BLENDER. For wet and dry mixing of powders, paint, flour, soaps, cosmetics, etc. All-welded construction; leak-proof packing gland. Sizes from 100 to 5000-lb capacity. D. B. Lewis Co., Los Angeles.

COMBINATION SHEAR AND PRESS. For shearing, punching, forming, and bending sheet metal. Capacity, 30 tons; 2-in. stroke; 100 strokes per min.; motor, 1½ hp, 3-phase, 1800 rpm; floor space, 36 by 75 in. Parker Mfg. Co., Santa Monica, Calif.

Processing

MAGNETIC SEPARATORS. Pulley-and-belt type with Alnico magnet pulleys. Provide automatic removal of tramp iron from wide variety of materials. Installed at discharge chutes, hoppers or conveyor belts. Belts 12 to 60-in. wide; any length. Dings Magnetic Separator Co., Milwaukee 14.

VIBRATING FEEDER. Electric; enclosed, dust-tight power unit. For use where materials are magnetic, hot, dusty, and abrasive. Slope of feeder adjustable from 5 degrees upgrade to 15 degrees downgrade. The Jeffrey Mfg. Co., Columbus 16, O.

Quarry

CRUSHER. Gyratory type. Centralized lubrication; overload protection against uncrushable materials; all-steel frame; forged steel shafts; alloy bronze bearings; quick, positive adjustment of crushing faces. Straub Mfg. Co., Oakland 7, Calif.

Textile

REDRAW MACHINE. Enclosed drive. Gate, revolving, or washer-type tension. Magnesium traverse rod adjustable for different-size bobbin heads. Self-threading Al-SiMag guides in bakelite holders. Sipp-Eastwood Corp., Paterson 4, N. J.

PRECISION TESTER. For fabrics, fibres, plastics, and similar materials. Screw loading system coupled to variable-speed motors by Thy-matrol control. Baldwin Locomotive Works, Philadelphia.

INSPECTING MACHINE. For measuring and examining. Variable-speed drive. 64-in. roller handles all materials to 60 in. wide. B. J. Phillips Co., N. Y.

FLEXIBLE
yet sturdy

Ward Leonard Edgeohm High Current Resistors for intermittent or continuous duty

Flexible in range, current and size, Ward Leonard Edgeohm Resistors are used in single or multiple assemblies. They are used for intermittent duty in motor starting, field discharge and dynamic braking in d-c controllers, and for continuous duty in high current battery charging rheostats, load banks, plating and welding.

Ward Leonard resistors are "Result-Engineered". By modifying a basic design, Ward Leonard can often give you the results of a special . . . for the price of a standard.

Write for Resistor Catalog. Ward Leonard Electric Co., 58 South Street, Mount Vernon, N. Y. Offices in principal cities of U. S. and Canada.

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WARD LEONARD
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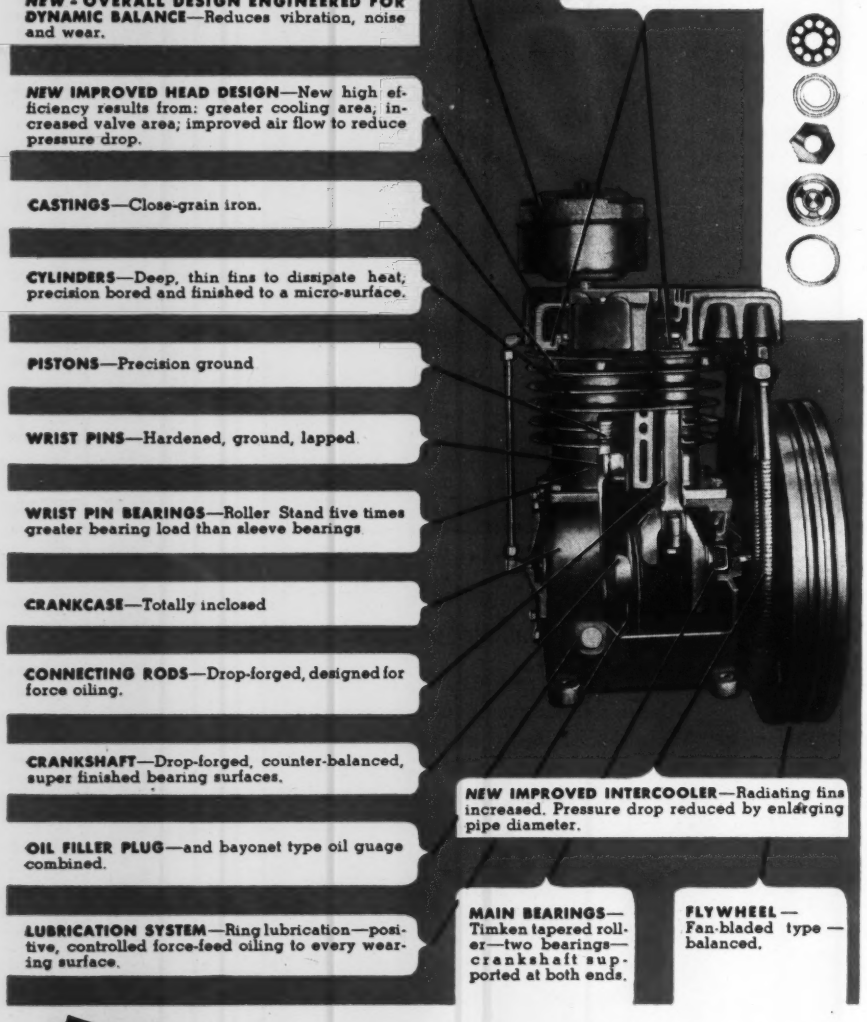
Why Kellogg-American More Air per Dollar

MEANS

A MORE EFFICIENT, ECONOMICAL

COMPRESSED AIR SYSTEM FOR YOUR MACHINE



NEW INTAKE — AIR-MAZE Unimaze Filter-Silencer. Highly efficient filtering of air and elimination of noise.

MULTIPLE VALVES— Light wafer-type valves with removable seats used for both inlet and outlet. Only two sizes of valves take care of all requirements — assure high efficiency and long life. Valve assembly consists of only five parts.

NEW - OVERALL DESIGN ENGINEERED FOR DYNAMIC BALANCE—Reduces vibration, noise and wear.

NEW IMPROVED HEAD DESIGN—New high efficiency results from: greater cooling area; increased valve area; improved air flow to reduce pressure drop.

CASTINGS—Close-grain iron.

CYLINDERS—Deep, thin fins to dissipate heat, precision bored and finished to a micro-surface.

PISTONS—Precision ground

WRIST PINS—Hardened, ground, lapped.

WRIST PIN BEARINGS—Roller Stand five times greater bearing load than sleeve bearings

CRANKCASE—Totally inclosed

CONNECTING RODS—Drop-forged, designed for force oiling.

CRANKSHAFT—Drop-forged, counter-balanced, super finished bearing surfaces.

OIL FILLER PLUG—and bayonet type oil guage combined.

LUBRICATION SYSTEM—Ring lubrication—positive, controlled force-feed oiling to every wearing surface.

NEW IMPROVED INTERCOOLER—Radiating fins increased. Pressure drop reduced by enlarging pipe diameter.

MAIN BEARINGS— Timken tapered roller—two bearings—crankshaft supported at both ends.

FLYWHEEL — Fan-bladed type — balanced.

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KELLOGG DIVISION

DESIGN ABSTRACTS

The Hydrosphere

IT HAS recently been discovered that a fitted spherical bearing is capable of hydrodynamically supporting loads of very great magnitude. The two components of this bearing are fitted together without clearance, the journal being a portion of a perfect sphere. The bearing surface is divided into a number of sectors by a multiplicity of equally spaced radial oil grooves. It is evident that when this bearing is subjected to an axial load there is no provision for the formation of a wedge in the direction of motion of the moving surface. Hence the source of the hydrodynamic pressure which is experimentally found to develop in this bearing is not readily apparent from our ordinary conception of lubrication.

Has Crescent-Shaped Film

A bearing having a wedge-shaped film in a direction normal to the motion of the journal is capable of acting hydrodynamically, provided there is flow within the bearing in a normal direction. However, the load capacity of any bearing is known to decrease rapidly as the flow from the bearing in the normal direction increases. The crescent-shaped film of the hydrosphere fortunately takes care of these opposing requirements by allowing appreciable flow within the bearing for the establishment of pressure, but a minimum of flow from the bearing. The seal which exists at the equator of the hydrosphere effectively limits the "end leakage" from this bearing. The flow from the equatorial region of the hydrosphere is indeed negligible and hence the operating temperature may be adjusted by merely proportioning the grooves to provide sufficient flow to carry away the heat generated. The oil-supply pressure may likewise be used to alter the flow of cooling oil as bearing operation is insensitive to oil-outlet pressure.

Like the Kingsbury and Michell bearings the hydrosphere does not depend upon a machine-generated surface to provide the minute inclination of the fluid wedge. The Kingsbury and Michell bearings require a pair of perfectly mating plane surfaces whereas the hydrosphere requires a pair of perfectly mating

Kodak

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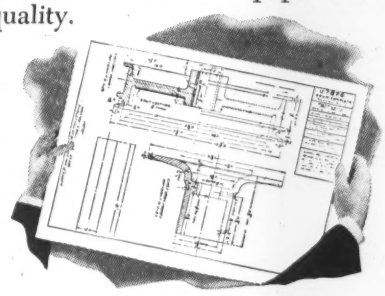

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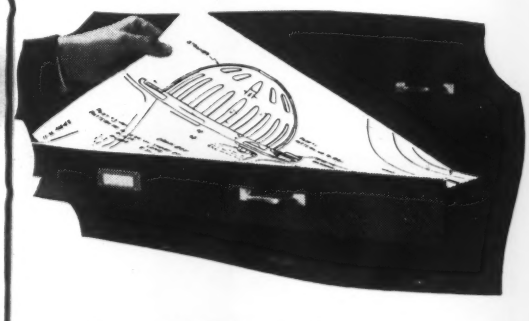

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FOR 25 years or more steels with an alloy content of 18% chromium and 8% nickel have shown such remarkable resistance to corrosion that they have earned the name of "Stainless" steels.

Although basically the same, the modern, 1948 models of the "18 and 8" stainless steels have been given a lot of PLUS values by recent metallurgical experience.

The analysis of Lebanon's Circle L 22XM (left) is a typical example. In addition to increases in the percentages of chromium and nickel, approximately 2.25% of molybdenum is added. It enhances the general corrosion resistance to many chemicals and is an important PLUS value to many manufacturers—perhaps to you.

If corrosion is one of your troubles, your metallurgists and design engineers should have a complete file of the new Lebanon Data Sheets describing these alloys. Write for them today.

LEBANON CIRCLE L 22XM

NOMINAL ANALYSIS

Carbon Max.	0.07
Silicon	1.25
Manganese	0.75
Chromium	19.50
Nickel	10.00
Molybdenum	2.25

NOMINAL PHYSICAL PROPERTIES

Tensile Strength	82,000
Yield Point	42,000
Elongation in 2"—%	50
Brinell Hardness	160

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"In The Lebanon Valley"
ORIGINAL AMERICAN LICENSEE GEORGE FISCHER (SWISS CHAMOTTE) METHOD

LEBANON ALLOY AND STEEL **Castings**

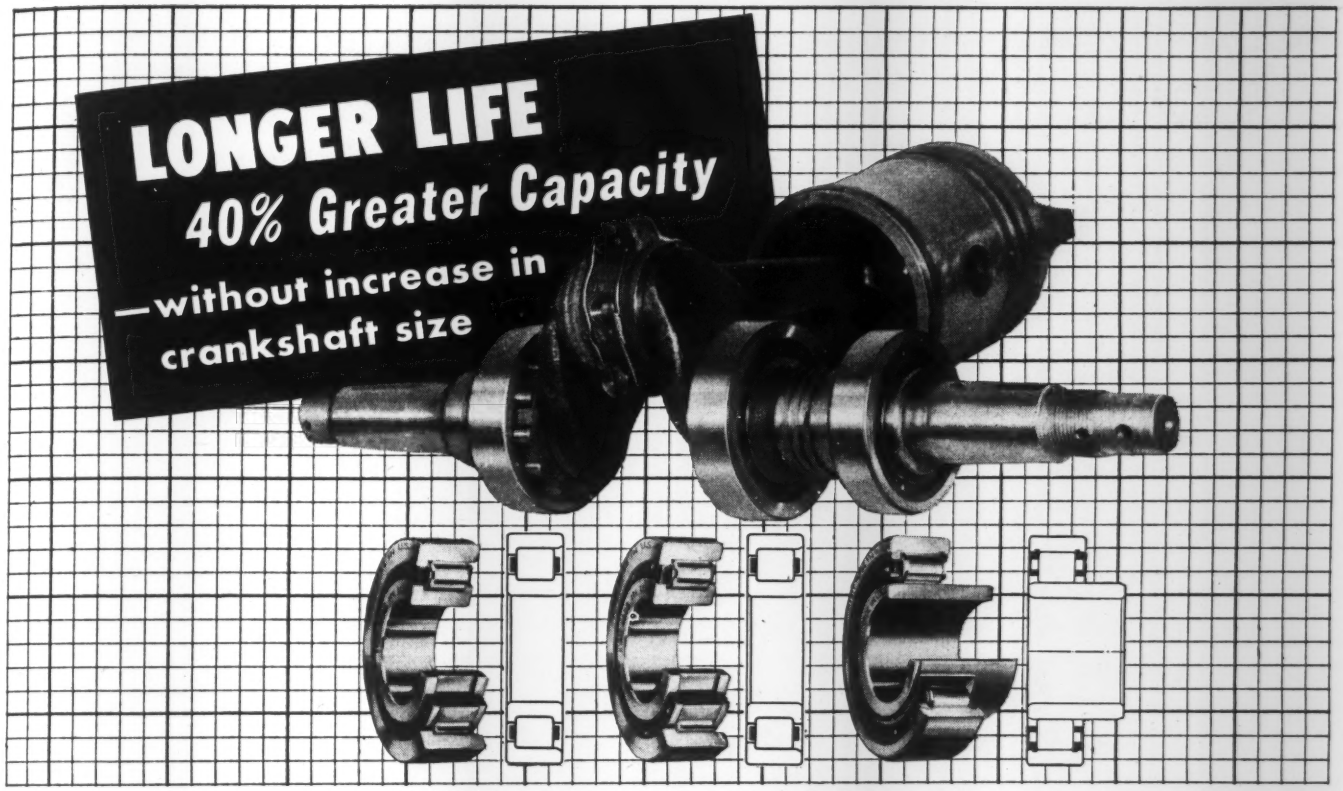
CIRCLE L

spherical surfaces. The precision obtainable by scraping in the case of the plane surfaces and by pressing and lapping in the case of the fitted spherical surfaces far exceeds that obtainable by the use of machine tools and ordinary measuring instruments.

The hydrosphere is angularly self-aligning and may operate in either direction of rotation when bath-oiled or pressure fed. The bearing has performed satisfactorily under a combined radial-thrust load. The relatively small effective radius of the hydrosphere (which gives a low friction torque) makes it of particular interest for high-speed applications. In this connection, it is to be expected that the stabilizing influence of the thin-film equatorial region of the bearing will prevent occurrence of the high-speed, self-excited vibration phenomenon known as "shaft whirl."
—From a paper by M. C. Shaw and C. D. Strang, Jr., Massachusetts Institute of Technology, presented at the recent annual meeting of the ASME.

Cost Reduction and the Designer

IT HAS been generally concluded that mass production is the answer to low prices. This is only partially true as has been demonstrated by automotive and other mass-production industries since the war. It is reasonable to assume that if a modern personal airplane were completely designed for mass production and placed on the market at substantially lower prices than are now prevalent, the increased sales would fall far short of defraying the expense of redesign and tooling. However, by clever redesign and by tooling in proportion to the market now available, it should be possible to encourage helathy growth of this market while operating on a profitable basis. The problem, then, is to analyze the overall picture of design, tooling, fabricating, and sales and determine the course which will best insure maximum realization of the objectives and benefits of mass production without all of the exorbitant costs of establishing a mass producing system. If we can achieve with relatively simple tooling and facilities only 50 per cent of the economies inherent in a true mass-production system, we can reduce the price to a range which will attract a greater number of owners who can profitably utilize conventional personal aircraft. This larger market will, in its turn, support more advanced tooling and production systems and also the cost of re-



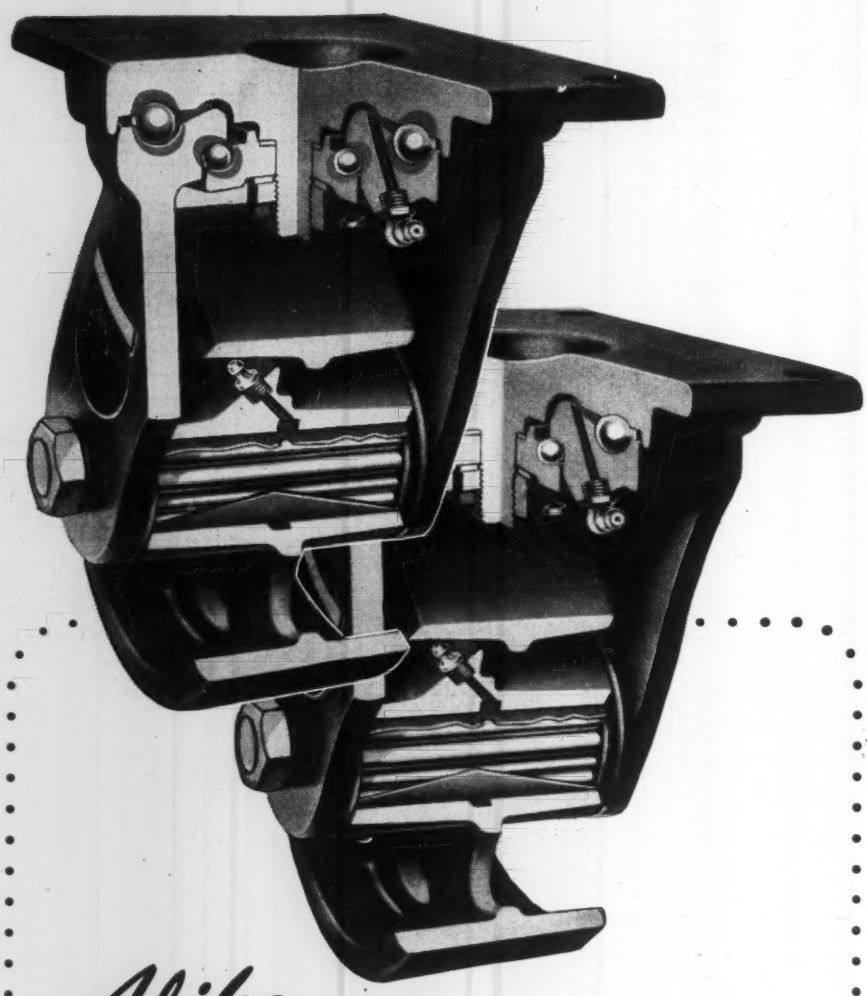
Without increasing the fundamental dimensions, Fairmont Railway Motors, Inc., designed 3 Hyatt Hy-Load Bearings into the crankshaft of their types RQ and RO engines. In addition to longer life and better performance, their tests over a three year period proved that the crankshaft "has a greater load capacity than can be attained by the use of any other kind of bearing." And the simplified construction of the bearings reduces engine maintenance, because servicing of the car can be done in the field instead of in the shop.

When machine designers—and Hyatt Hy-Load Bearings—get together, you can expect results like this.

The Hyatt Hy-Load line, with *ten* major types of cylindrical roller bearings, offers designers greater flexibility in design possibilities. It helps production men simplify and speed up assembly. And because these bearings are designed right—made right—they mean BETTER PERFORMANCE wherever they are used. Hyatt Bearings Division, General Motors Corporation, Harrison, N. J., Chicago; Detroit; Pittsburgh; and Oakland, Calif.

FOR THE TOUGH LOADS USE HY-LOADS

HYATT ROLLER BEARINGS



Alike...

until friction goes to work

Left: A Rapids-Standard Steel-Forged Caster. Right: The same caster as we *might* make it. The difference...not important until friction, shock and heavy loads go to work on the bearing raceways. Then the difference stands out. Only in the Rapids-Standard caster are these raceways flame-hardened...given the extra strength and hardness to stand up under years of the toughest service.

Add a steel-forged top plate and swivel yoke, a high carbon steel axle of heavy-duty size, Hyatt or oil impregnated wheel bearings, a choice of wheels (Nicro-steel, rubber or plastic), a special fitting that eliminates side thrust wear...and that's the Rapids-Standard Steel-Forged Caster.

THE RAPIDS-STANDARD CO., INC.
330 Rapistan Bldg., Grand Rapids 2, Mich.
Representatives in all principal cities

Rapids-Standard

STEEL-FORGED CASTERS

search and development on new design.

When we stop to analyze all of the variables which contribute to airplane costs, it is interesting to note the number of such variables directly or indirectly related to the efficiency of our Design Engineering. Design Engineering encompasses more than purely technical considerations. It extends into certain phases of almost every function of a modern manufacturing organization; hence, it calls for closest coordination and cooperation with all of the various departments. Knowledge of current problems, activities, and opinions of Purchasing, Material Control, Tooling, Production, Inspection, Sales, and Service personnel is essential to the development of efficient design. It has been said that there are innumerable ways to "skin a cat"; hence, we will, for the moment, disregard primary design problems and assume that a workable technical solution is available in each case. Let us turn to an enumeration of some of the most important practical nontechnical factors which must be considered in any design program directed toward maximum production economy:

1. Material cost and availability in various forms (including purchased parts)
2. Type of equipment available (and relative utilization)
3. Type of personnel and skills required
4. Nature and extent of tooling required
5. Probable number of units to be produced
6. Inventory on hand subject to use or obsolescence
7. Number of pieces and parts involved in design
8. Capital available to underwrite costs
9. Special requirements established by management.

In initiating a new design where production cost is a factor, no variable is too insignificant for consideration. The general problems should all be investigated and evaluated at the outset before any attempt is made to begin the actual design. Consideration of current prices of all raw materials in all available forms will come under this general classification.

If current price listings are all assembled for evaluation, comprehensive comparisons can be made in determining the best course for realization of maximum production economy. This information may influence

"Here's Why I Specify BRISTO MULTIPLE-SPLINE SOCKET SCREWS"

DESIGNED FOR MORE *HOLDING POWER.* The multiple-spline principle is one of the most efficient methods of transmitting torque. In socket screws, the multiple spline directs all the force applied in "wrenching up" into *rotary motion.* For this reason, multiple-spline screws have great inherent strength, can be tightened far beyond the point where hex-type screws round out or burst. Result: tighter grip . . . greater holding power.

RESISTS SEVERE VIBRATION. The Bristo Multiple-Spline Socket Screw is ideal where severe vibration is a problem . . . disassembly frequent . . . fastening points hard to reach . . . or where small size presents a breakage hazard.

PRECISION THREADING. Bristol precision, famous for over 50 years in the instrument field, is strongly reflected in Bristol screw products. Careful design, close tolerances and precision threading, are backed up by a complete inspection system. This includes close control over alloys, testing the final product for hardness and ultimate strength, gauging threads for lead and pitch and ending with 100% visual inspection before packing to insure a *precision* socket screw. Result: quicker, easier fastening on the job . . . lowered assembly costs.

"INTERNAL WRENCHING" GIVES DESIGN ADVANTAGES. Socket screws require less pocket allowance in the product for wrenching . . . allow reductions in flange sizes . . . permit neater, more compact design.

BRISTOL HEX SOCKET SCREWS. Bristol Hex Socket Screws are made from the same materials . . . by the same processes . . . to the same specifications as the Bristo Multiple-Spline, differing only in socket design. They give superior service in ordinary socket screw applications.

Write for bulletin and prices. THE BRISTOL COMPANY, Mill Supply Division, 122 Bristol Road, Waterbury 91, Conn.

ONLY THE "B-LINE" HAS THE RIGHT SOCKET SCREW FOR EVERY APPLICATION

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MILL SUPPLY DIVISION

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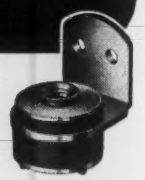
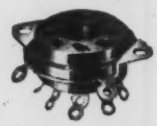
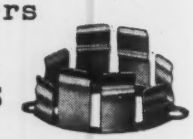


in quantities suitable to Volume Production...it may pay you to call upon the Design Engineers of United-Carr and its



subsidiaries. They have helped many manufacturers



★ CUT COSTS
★ SPEED PRODUCTION
★ TURN OUT FINER FINISHED PRODUCTS

UNITED-CARR FASTENER Corp.
CAMBRIDGE 42, MASSACHUSETTS

MAKERS OF DOT FASTENERS

the final decision as to whether the component will be cast, stamped, forged, extruded, or formed from sheet stock. Cost comparisons of the various material forms can be of considerable value in this respect. Also, the indirect problem of balancing cost against weight can be solved on a sounder basis when all the facts regarding relative cost of various materials is available.

Using Equipment on Hand

Now that the type and amount of production equipment in most aircraft plants are more stabilized than they have been in the past, efficient use must be made of equipment on hand, unless, of course, there is justification for further procurement. To do this efficiently, we must assemble cost breakdowns that are complete in that they include amortization of the original cost of the equipment as well as the maintenance and operational expenses involved. I believe it has been or is the practice of most companies to charge off the first two of these items to overhead without consideration of cost breakdowns, a policy which can obviously lead the designer to false conclusions of economy. Caution must be exercised in considering exact breakdown figures, because overloading of equipment showing the least operational costs can result unless close contact is maintained with the Planning department on schedules.

Shop Personnel and Design

Even the subject of personnel enters the picture in designing for economy. Current labor conditions are reflected in the various cost breakdowns, but the type of personnel to be involved is peculiar to the locality and may establish trends toward which many of the components should be designed. For instance, in the East skilled machinists are more plentiful than in other sections of the country, and knowledge of this fact will influence a design program accordingly.

Next, there are tooling considerations. The size of the tool design staff and the amount of tool and die equipment and skilled personnel available will all have their effect on the design program. If only a short time can be allocated to tooling the project, then due consideration must be given to temporary as well as long-range tooling. The extent of the tooling program is, of course, directly connected with the total probable number of units to be produced, and

U·S·AUTOMATIC CORP.
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Rely on this trade-mark for precision screw machine parts.

BAR CAPACITY 3/64" to 5 1/2"

More than a hundred automatic screw machines are ready to make your parts now.

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Wide inner ring ball bearing with exclusive self-locking collar for slip-fit assembly and disassembly
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Wire spring in base for grounding electrical charges
Inch-dimension bore sizes up to 1 7/16" for popular shaft sizes.
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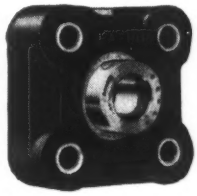
Noise and Vibration
FOR LIGHT-DUTY
Fafnir Rubber-Mounted Ball Bearing Units



. . . also rubber mounted flange and cylindrical cartridges

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